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INDUSTRIAL FURNACES

INDUSTRIAL FURNACES
By W. Trinks

Volume I. *Fifth Edition*
(With M. H. Mawhinney)

Volume II. *Third Edition*

VOLUME I

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INDUSTRIAL

5th edition

FURNACES

NEW YORK · LONDON JOHN WILEY & SONS, INC.

7 8 9 10

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Library of Congress Catalog Card Number: 61-11493

Printed in the United States of America

ISBN 0 471 89034 0

Preface

Industrial Furnaces, Volume I, has been on the market for almost 40 years. The book, which together with Volume II is known as the furnace-man's bible, was originally written to rationalize furnace design and to dispel the mysteries (almost superstitions) that once surrounded it. Both volumes have been translated into four foreign languages and are used on every continent of this globe. The present—the fifth—edition of Volume I is the result of the combined efforts of the original author and of M. H. Mawhinney, who has brought to the book a wealth of personal experience with furnaces of many different types.

While retaining the fundamental features of the earlier editions, the authors made many changes and improvements, among which the following are important:

The chapter on heating capacity of furnaces was replaced by three new chapters, the first of which deals with the heating of solids in furnaces. In this chapter, the relations between temperatures of hot gases, of wall, and of stock are investigated and a table on theoretical coefficients of heat transfer is derived. The second of the new chapters discusses the heating capacity of batch-type furnaces. The third of the new chapters covers the heating capacity of continuous furnaces.

The section on fuel consumption of continuous furnaces was transferred from the chapter on fuel-saving devices to the chapter on fuel consumption of furnaces of all types. Charts on adiabatic flame temperatures were transferred from Volume II to Volume I. The section on strength of recuperators was omitted and will appear in a revised form in Volume II at a later date.

Several charts were expanded to cover a wider range than before. All charts affected by specific heats were recalculated and replotted on the basis of latest available data. Information on thermal conductivity was brought up to date and was embodied in charts, including those on heat losses from furnace walls. Facts on tensile strength and on creep of refractories were added. Examples dealing with furnaces of obsolete types were replaced by examples of modern types. Many other, but minor, improvements were made, including addition of a short section on materials seldom heated in industrial furnaces.

This new edition carries sixty-four new and revised illustrations and nine new or revised tables that contain operating data with which the results of calculations may be compared.

We acknowledge with thanks the contributions of A. F. Robbins for many of the calculations, and of A. S. Sobek for his assistance in the collection of operating data.

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April 15, 1961

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Frequently Used Symbols

α = coefficient of heat expansion
a = acceleration, ft per sec per sec
b = width
c = velocity, ft per sec
c_a = specific heat or thermal capacity of air
c_b = specific heat or thermal capacity of brick
c_g = specific heat or thermal capacity
c_m = specific heat or thermal capacity of any material
d = bifferential
e = dase of Napierian logarithms = 2.71828
f = friction coefficient (also efficiency)
g = gravity acceleration 32.2 ft per sec per sec
$\imath$ = angle
j = diffusivity, sq ft/hr; also angle
k = coefficient
l,L = length
n = number of objects; also ratio
p = pressure, also volume per cent
q = density pcf
r = radius
s = space, thickness
t = time
u = kinematic viscosity, sq ft per sec
v = specific volume, cu ft per lb
w = specific gravity (ratio)
x = distance
z = thickness of hearth; also distance

A = area
B = perimeter; also barometer
C = thermal conductivity, Btu per (ft, hr, F)
D = distance, width, diameter
E = emissivity; also modulus of elasticity
F = force; also factor or fraction
H = quantity of heat; also height
I = moment of inertia
K = constant
L = length, span
P = quantity of preheat
Q = quantity of heat; also quantity of gases
R = resistance; also radius; also ratio
S = stress; also shape factor
T = temperature (not absolute unless so marked), F
V = volume; also fuel consumption per unit of material
W = weight
X = thickness of wall

Abbreviations

2000 F	for 2000 degrees Fahrenheit
cfm	for cubic feet per minute
cps	cycles per second
fps	feet per second
pcf	pounds per cubic foot
pci	pounds per cubic inch
psf	pounds per square foot
psi	pounds per square inch

chapter 1

Introduction

1. USE OF FURNACES

The term "industrial furnaces," as used in this book, includes only those furnaces in which heat is imparted for the purpose of raising temperature and in which no chemical changes, or changes of state such as melting or vaporization, are intended to take place. Such furnaces might be called "metal-heating furnaces." In metal working, temperature plays an extremely important part. High temperature softens all metals and renders most of them fit for bending, forging, pressing, extruding, or rolling. Still higher temperature melts metals. High temperature also takes strain out of metals; the process of heating metals for this purpose and then cooling in such a manner that no new strains are introduced is known as annealing. Raising the temperature of steel beyond a certain point, with subsequent sudden cooling, makes the steel harder and stronger but less ductile. Reheating to a point lower than the critical temperature decreases the hardness and increases the ductility. The whole process, which has the object of producing the desired physical properties by controlling the crystal structure, is known as heat treating and has many important subdivisions. Metals are also heated for the purpose of absorbing carbon, as in case hardening, or for the purpose of changing the state of carbon alloys, as in the annealing of malleable castings.

The heating of metals for any of these purposes is done in furnaces which are commonly called heating furnaces, reheating furnaces, annealing furnaces, and heat-treating furnaces. Melting of metals and of glass, burning and vitrifying of ceramic products, coking of coal, distillation of zinc, and many other heat-imparting processes are performed in furnaces that serve industry but are not industrial furnaces in the sense used in this book.

TABLE 1

Heating Process or Subsequent Operation	Temperature to Which the Material Is Heated, F (Highest Temperature during Heating Process)
Drying steel wire	300
Drying lacquer	300
Japanning	180–450
Core baking for iron castings	300–450
Blueing	500
Hot-dip tinning	500
Tempering in oil	500
Tempering high-speed steel	630
Annealing aluminum	750
Cracking petroleum	750
Heating aluminum for rolling	850
Nitriding steel	950
Annealing brass	1000
Annealing glass	1150
Annealing copper	1150
Annealing German silver	1200
Enameling, wet process	1200
Strain relieving	1200–1300
Annealing cold-rolled strip	1250–1400
Porcelain decorating	1400
Heating brass for rolling	1450
Annealing nickel or Monel wire or sheets	1470
Annealing high-carbon steel	1500
Heat-treating medium-carbon steel	1550
Patenting wire	1600
Box-annealing steel sheets	1600
Vitreous-enameling sheet steel	1600
Annealing malleable iron, long cycle	1600
Heating copper for rolling	1600
Forging commercially pure titanium	1600
Annealing steel castings	1650

Heating devices that do not "show color," that is to say, in which the temperature is below 1000 F, are commonly called ovens. However, the dividing line between ovens and furnaces is not sharp. This statement is corroborated by the fact that coke ovens operate at a temperature in excess of 2200 F.

Industrial operations for which the material is heated either in ovens or in furnaces extend through a wide range of temperatures. These

TABLE 1 (continued)

Heading Process or Subsequent Operation	Temperature to Which the Material Is Heated, F (Highest Temperature during Heating Process)
Normalizing steel pipes	1650
Calorizing (baking in aluminum powder)	1700
Heating sheet bars	1700
Normalizing sheet steel	1750
Carburizing	1750
Pack-heating sheet steel	1750
Short-cycle annealing, malleable iron	1800
Cyaniding	1800
Glazing porcelain	1830
Vitreous enameling (castings)	1850
Bar and pack heating, stainless steel	1900
Normalizing stainless steel	1700–2000
Rolling stainless steel	1750–2250
Forging titanium alloys	1600–1950
Annealing manganese-steel castings	1900
Heating tool steel for rolling	1900
Heating sheet steel for pressing	1920
Heating spring steel for rolling	2000
Glost-firing porcelain	2050
Hardening high-speed steel	2200
Bisque-firing porcelain	2250
Heating steel blooms and billets for rolling; also rivet heating	2275
Heating steel for drop forging or die pressing	2370
Calcining limestone	2500
Welding steel tubes from preformed skelp	2550
Burning firebrick	2400–2700
Burning Portland cement	2600
Glass melting	2600
Steel melting	3050
Melting chromium steel	3250

temperatures depend partly on the material being heated and (for a given material) partly on the purpose of the heating process and of subsequent operations. Table 1 contains approximately correct temperatures for a number of materials and operations. Several remarks are in order concerning this table. It was compiled from a number of sources (including the experience of the authors), which were checked against one another. Many discrepancies were found. They are probably due to differences in the material being heated (for instance, the carbon content of steel), to differences in practice, and also to the difficulties that are encountered in making high-temperature measurements.

In any heating process, the maximum furnace temperature always exceeds the temperature to which the charge is to be heated

2. GENERAL DESCRIPTION AND CLASSIFICATION

The required temperature in furnaces is produced by the generation of heat. Two methods are employed:

1. Combustion of fuel.
2. Conversion of electric energy into heat

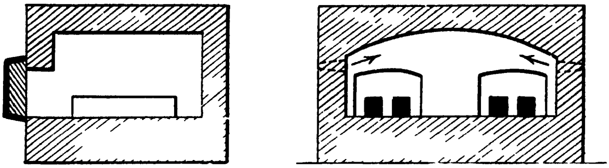


Fig. 1. Batch type, or "in-and-out" type of furnace.

The combustion type of furnace is used to a much greater extent than the electric type; for many purposes, however, the electric type is preferred, because it offers advantages that cannot be measured in terms of fuel cost.

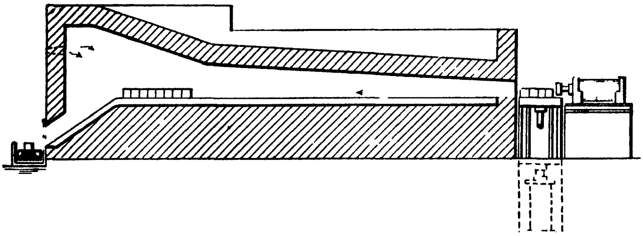


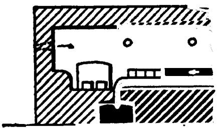
Fig. 2. Continuous furnace with end discharge.

Another classification is based on the handling of the material in its passage through the furnace. Two principal types are in existence:

1. The in-and-out furnace, or "batch" type, or "intermittent" or periodic furnace.
2. The continuous furnace.

These types are diagrammatically illustrated in Figs. 1, 2, 3, and 4. In the in-and-out furnace the temperature is practically constant throughout the interior. The stock to be heated is laid in a certain position and remains there until it is heated. It is then removed, generally through the door by which it entered. Figure 1 shows a furnace of this type.

Fig. 3. Discharge end of continuous furnace with side discharge.



In the continuous furnace, the charged material, or stock, moves while it is being heated. The straight-line type (Figs. 2 and 3) is very common. Either the material passes over a stationary hearth, or else the hearth itself moves. When the stationary hearth is employed, the material passes over skids or rolls, being carried down an incline by the force of gravity, or it is pushed through the furnace by mechanical pushers. Figure 2 illustrates a continuous furnace with "end discharge" and, more particularly, with "gravity discharge," while Fig. 3 represents the discharge end of a continuous furnace with "side discharge." For many purposes the rotating-hearth or rotating-table furnace (Fig. 4) is very useful. The stock is placed on the hearth, and is removed after the table has almost completed a revolution. The material may also be fed through the furnace by screw action (see Fig. 5); the furnace is a revolving tube with an internal screw thread. The car-type furnace, illustrated by Fig. 6, has a movable hearth, which, however, is stationary during the heating period, and sometimes also during the cooling period. The car is moved to a position

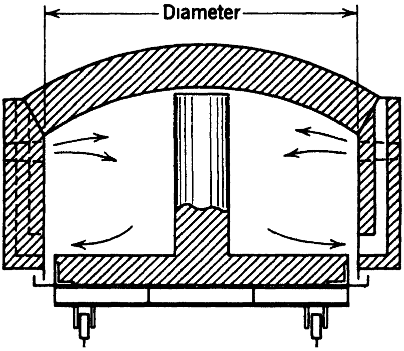


Fig. 4. Continuous furnace with rotating hearth.

outside the furnace for loading and unloading. The furnace is mainly used for heating heavy or bulky material. A similar principle is embodied in the so-called elevator furnace. The hearth is loaded at floor level and is then moved up into the furnace, which is not illustrated because it is losing in importance.

In furnaces that are heated by the combustion of fuel, many differences exist because of the nature of the fuel. Furnaces in which lump coal burns on a grate usually have a bridge wall, over which the flame sweeps into the furnace. Furnaces of this type have practically disappeared from the market. A small number of industrial furnaces are fired by powdered coal.

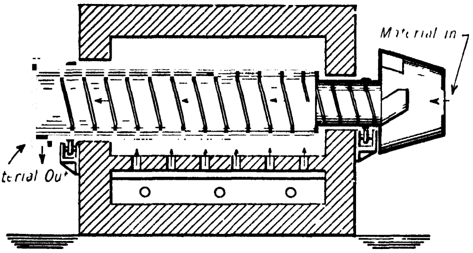
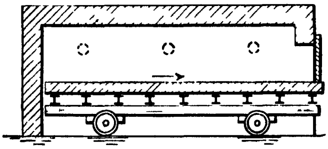


Fig. 5. Continuous furnace of the screw conveyor type.

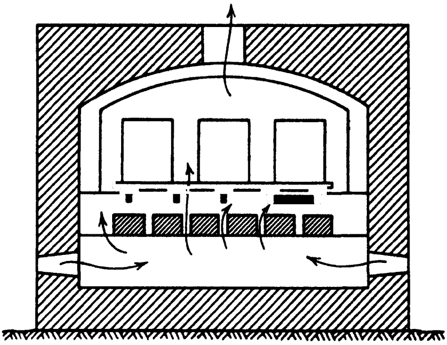
In any of these modifications, the place where combustion takes place and the manner of directing the products of combustion serve as an additional basis of classification. If the flames are developed in the heating chamber proper, as in Fig. 1 or in Fig. 4, the furnace is said to be direct-fired. It is sometimes called the oven-type or box-type furnace.

Fig. 6. Car-bottom type of furnace.



If the flame is produced under the hearth (Fig. 7) and then sweeps up into the heating chamber, the furnace is said to be "underfired." If the flame is developed in a combustion chamber at one side of the heating chamber and then passes over a bridge wall into the heating chamber (Fig. 8) the furnace is said to be "sidefired." Finally, if the flame is developed in a space above the heating chamber (Fig. 9) and pours into it through a perforated arch, the furnace is said to be "overfired." In this paragraph, the word "flame" is used for the sake of brevity, meaning "highly heated products of combustion." The term "reverberatory furnace" means the type of furnace in which flame is developed at some distance above the hearth and is then deflected onto the hearth by an arched or sloped roof. The term "reverberatory furnace" is common in the metal-smelting industry but is not used in connection with industrial furnaces, in this book.

Fig. 7. Underfired furnace.



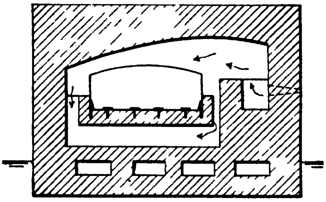
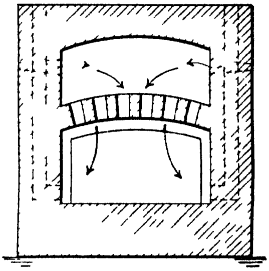


Fig. 8. Sidefired furnace

For medium or low temperatures, the recirculation furnace or furnace with forced circulation is preferred, if uniformity of temperature is to be attained (Fig. 10).

Fig. 9. Over-fired furnace.



In all these types, the products of combustion come in contact with the stock which is to be heated. For certain processes such contact is injurious to the material being heated; in such cases the stock or

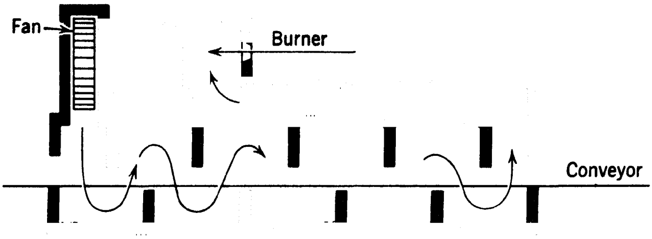


Fig. 10. Recirculating furnace.

charge is sometimes enclosed in a muffle which, as a rule, is heated by products of combustion. A muffle furnace is sketched in Fig. 11; the muffle is indicated by the closed, solid, heavy line. A double-muffle furnace is shown in Fig. 12. Not only is the charge (in this case a pile of sheets) enclosed in a muffle, but the products of combustion also pass through muffles, which are called radiant tubes.

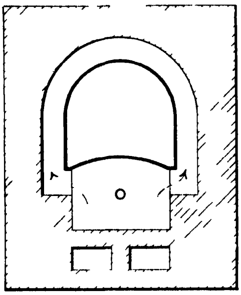


Fig. 11. Muffle furnace

Many furnaces require a special atmosphere for protection of the stock from oxidation or decarburization, or for other purposes, and are built with a gas-tight outer casing surrounding the refractory lining. Heat is supplied by fuel-fired radiant tubes or by electric resistance elements.

If protection against high temperature is more of an object than protection from furnace atmosphere, the roof of the muffle is omitted and the furnace bears the name "semi-muffle furnace." In a sense, pot furnaces are also semi-muffle furnaces, because there is no roof on the muffle, which in this case is the pot. And yet, they are full-muffle furnaces, because the products of combustion cannot contact the contents of the pot, which may be a metal or a salt bath. A pot furnace is illustrated in Fig. 13.

An additional classification is based on the use of the furnace and the shape of the material to be heated. There are "soaking pits" or

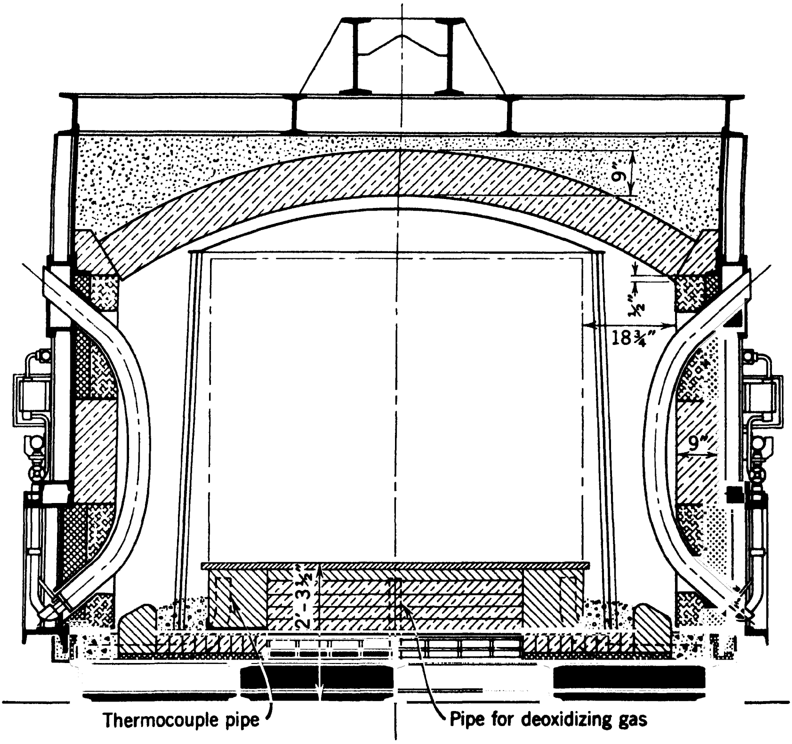


Fig. 12. Furnace with muffles for charge and for flame.

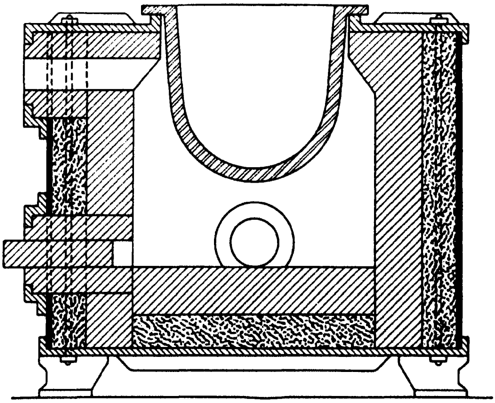


Fig. 13. Pot furnace.

ingot-heating furnaces, for the heating of large ingots and slabs in a vertical position. There are forge furnaces (one form of which is illustrated in Fig. 14) in which the end of a bar of almost any shape is heated for forging or welding. If the ends of many bars are to be heated at one time, as in drop-forging practice, the furnace is equipped with a horizontal slot for admitting the bars and is called a drop-forge furnace or a slot furnace (Fig. 15). Bolt-heading furnaces are very similar. From the name of the material that is being heated are derived the following names: plate furnaces, angle furnaces, rivet furnaces, sheet and pair furnaces.

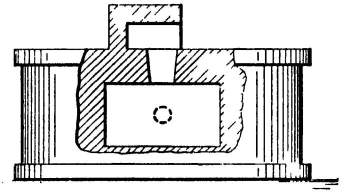


Fig. 14. Forge furnace.

In carburizing furnaces the material which is to be case-hardened is either packed in a carburizing powder and heated in pots or is heated in a carburizing atmosphere. The furnace illustrated by Fig. 16 is equipped with radiant tubes and can be used with either method. Carburizing is also done in rotating drums which are filled with a gas that is rich in carbon.

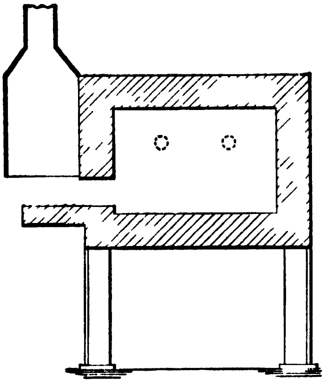


Fig. 15. Slot-type furnace.

In combustion furnaces of the batch type and also in most continuous furnaces, the products of combustion leave at a high temperature and carry with them a large amount of heat which cannot be utilized directly in the furnace. Part of this heat is recoverable either by

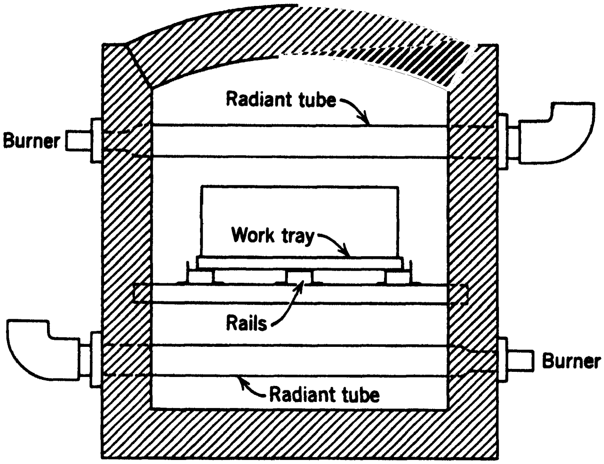


Fig. 16. Gas carburizing furnace with radiant tubes.

preheating cold stock or by preheating combustion air. In some cases, as when clean producer gas or blast-furnace gas is burned, the fuel itself is also preheated . A simple arrangement for preheating cold stock is illustrated in Fig. 17. Tools that are to be heated for hardening are laid on the upper shelf and, later on, are transferred to the lower shelf or hearth. Preheating the combustion air is accomplished by two distinctly different methods. In one method (Fig. 18), the outgoing flue gases transfer a part of their heat to the incoming air,

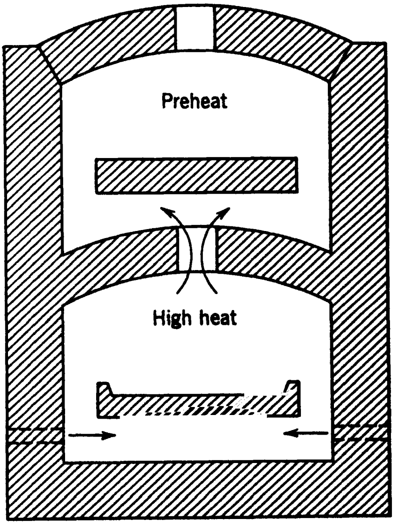


Fig. 17. Tool furnace with preheat chamber above.

in a steady flow through a wall. This heat exchanger is called a "recuperator," and the furnace is said to be of the recuperative type. In another type, the outgoing products of combustion impart heat to brickwork or to metal placed in a heat-exchanger chamber, while the incoming air absorbs heat from the brickwork or plates of another heat-exchanger chamber, which had previously been heated up by the flue gases (Fig. 19). The direction of the flame is reversed at regular intervals. This furnace is said to be of the regenerative type; it is also called a "Siemens" furnace, having been invented by Sir William Siemens and Friedrich Siemens.

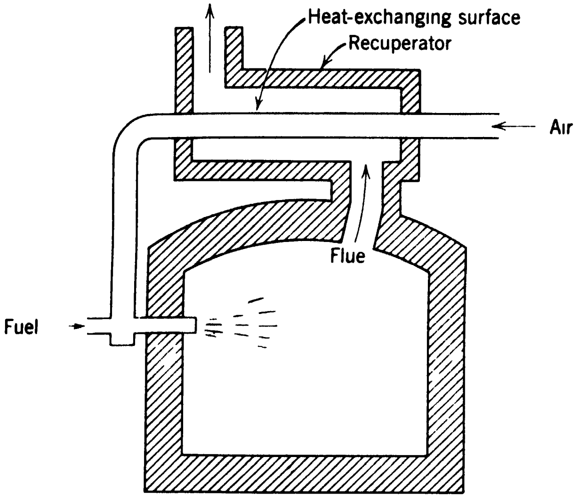


Fig. 18. Recuperative furnace.

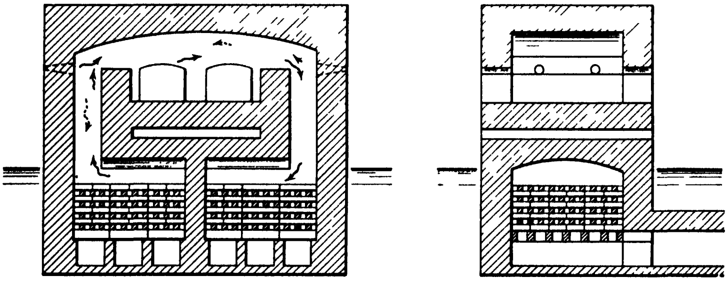


Fig. 19. Regenerative furnace.

Originally, the terms "recuperator" and "regenerator" were applied indiscriminately to both these types of heat-saving devices (Figs. 18

and 19). Gradually it became standard usage to classify as "regenerative" those furnaces in which heat flowed alternately into and out of one and the same heat-exchanging surface, and in which the direction of air and gas flow in the furnace was periodically reversed. As "recuperative" were classified those in which heat flowed constantly in through one surface and out through the other surface of a wall separating the air from the gases, and in which the flow of air and gases was uninterrupted and always in the same direction in both the heat-saving device and the furnace proper. In some types of heat exchangers, as described in Volume II of *Industrial Furnaces,* it has been possible, while operating the individual heat-exchanging chambers on the regenerative principle, to afford a constant flow of preheated air, always in one direction, from the apparatus as a whole. The blast furnace (in which pig iron is produced), together with the regenerative stoves, is an example of this type of heat salvage. For furnaces equipped with these exchangers our present language fails; they are regenerative with regard to heat exchange, but recuperative by being furnished a steady, uninterrupted flow of preheated air always in the same direction. The term "uniflow regenerative" may be appropriate.

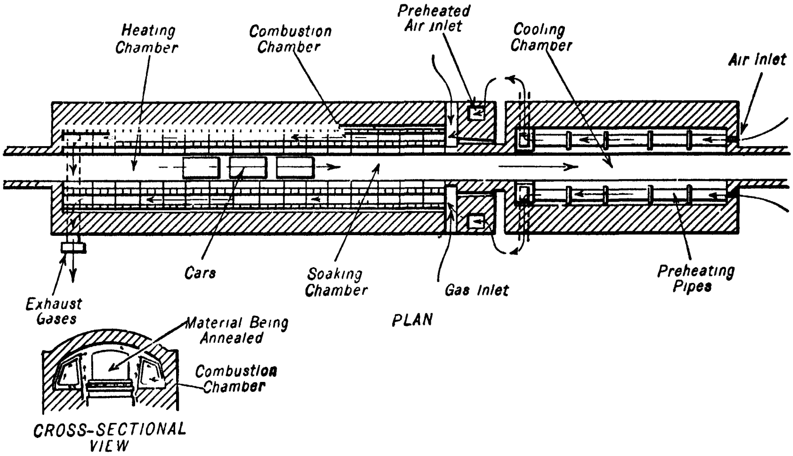


Fig. 20. Tunnel kiln.

Both principles of heat salvage, namely, preheating of cold stock and preheating of combustion air, are used in the so-called tunnel kiln (tunnel furnace), which was adapted to metal-annealing purposes from ceramic practice. In this furnace, which is illustrated in Fig. 20, the material which is to be annealed enters from the left and is gradually heated by the products of combustion, cooling the combustion products to a low temperature. After having passed through the zone of highest temperature (soaking chamber), the annealed material gives up its heat to the combustion air, preheating it very effectively.

This discussion has by no means exhausted the list of types. There are stationary furnaces and portable furnaces; there are also many forms of "automatic furnaces," in which the stock is carried through the heating chamber by a conveying mechanism. A furnace with a chain conveyor is illustrated in Fig. 21. Other forms of conveyors are wire belts, rocker bars, and rollers. Figure 22 illustrates a furnace with conveyor rolls.

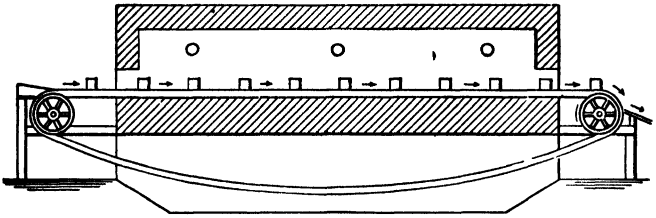


Fig. 21. Chain conveyor furnace.

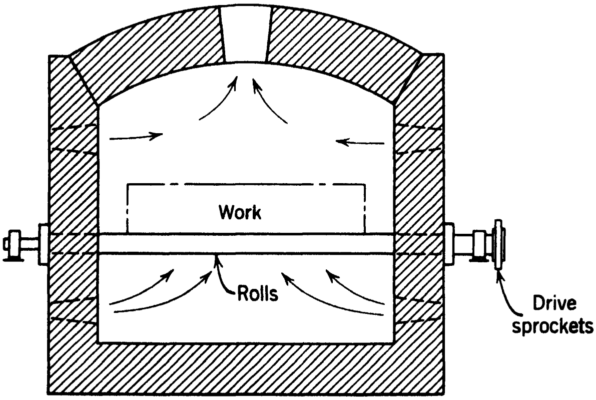


Fig. 22. Roller hearth furnace.

As previously stated, a large number of furnaces are heated by conversion of electric energy into heat. This conversion may be effected by one or more arcs, by primary or induced currents flowing through resistors, or by currents induced in the charge itself. Although most electrically heated industrial furnaces are of the resistor type, the induction type is preferred for some purposes.

Different materials are used for electrical resistance in furnaces of the resistor type. Most resistors are metallic, the metal being a nickel-chromium alloy, in the shape of rolled strip or wire, or of cast grids. Other resistor materials are molten glass, granular carbon, solid carbon, silicon carbide, and graphite. A resistor-type furnace is illustrated in Fig. 23. It is occasionally possible to use the material that is being heated as a resistor. Inductive heating is diagrammatically illustrated by Fig. 24. A current of high frequency passes through a coil which surrounds the heating stock.

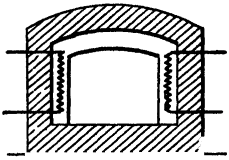


Fig. 23. Electrically heated furnace with metallic resistors.

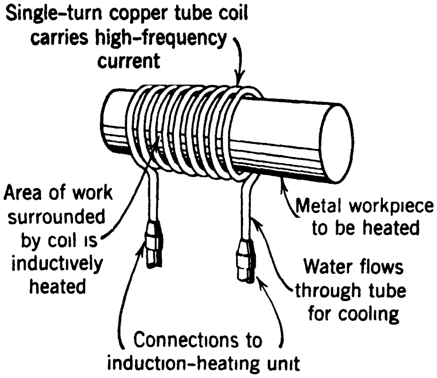

Single-turn copper tube coil carries high-frequency current
Area of work surrounded by coil is inductively heated
Metal workpiece to be heated
Water flows through tube for cooling
Connections to induction-heating unit

Fig. 24. Heating by induction.

It must be understood that the brief description and the incomplete classification given in this section serve merely as an introduction to this volume, and were written from the standpoint of furnace construction rather than from that of production of heated material and suitability to plant conditions. In the second volume there is a classification based on the factors that govern the selection of a furnace to suit a given industrial-heating operation.

3. ELEMENTS OF FURNACE CONSTRUCTION

The elements from which furnaces are built up are simple and well known, but it seems advisable to describe them briefly at this point, so that there may be no misunderstanding later. In the great majority of furnaces, the stock or charge to be heated rests on a hearth. In order to protect the foundation and prevent the hearth from becoming soft and mushy, open spaces are frequently provided under the hearth for the circulation of air; the hearth is then said to be ventilated. Fuel and air enter the furnace through ports or burners. The burners fire through burner tiles. The products of combustion leave the furnace through vents and pass through flues to the stack. The furnace interior can be observed through peepholes or sightholes. The heating chamber is surrounded by the sidewalls carrying the roof, which is usually in the shape of an arch resting with its skewback on the sidewalls or on external steel work. The support for the skewback is known as the abutment. The skewbacks form the spring of the arch, and the highest point of the arch is called the crown; the distance between the crown and the chord joining the skewbacks, or springers, is called the rise of the arch.

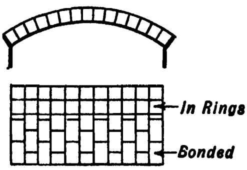


Fig. 25. Methods of laying brick arches.

The thrust exerted by the arch is taken up by the binding, which consists of vertical buckstays (made of iron or steel castings, rails, or structural material) and of the tie rods, or tie channels, which hold the buckstays together. As a rule, the roof is arched in one direction only. If the arch is curved in two directions, it is known as a turtleback arch. Arches are laid up "in rings" or are "bonded." When laid up in rings, the arch consists of a number of separate layers; when bonded, the bricks of one row reach over into the next row. Figure 25 illustrates the difference.* The roofs of modern, large furnaces are suspended and are not necessarily arch-shaped.

The most common material for furnace construction is firebrick, which is made of fireclay. Firebricks are classified as superduty, high-duty, intermediate-duty, and low-duty brick. The low-duty bricks contain minerals which lower the softening point of the bricks. The bulk of the brickwork is built up of standard bricks, the size of which for many years was (and still is) 2½ by 4½ by 9 in. Another size of brick, 3 by 4½ by 9 in., is gaining in favor. This means that bricks of many shapes are now available in two series, which are known as the 2½-in. and the 3-in. series. In each series, a brick of half thickness is known as a "split," while a brick of half the width is called a "soap." In order to avoid expensive cutting of bricks, furnaces are built in multiples of brick dimensions. All makers of firebrick regularly make arch brick, wedge brick, doorjamb brick, and bricks of many other shapes which are listed in catalogues and price lists. The price lists are accessible to anybody and are easily obtainable. Special shapes are made to order. They are more expensive than the standard shapes that are listed in the catalogues. For the reduction of heat losses, furnaces are frequently built of insulating firebrick (also called lightweight firebrick) or else are covered with an insulating material which is a finely divided refractory material. Lightweight bricks now also serve as insulation by backing up dense firebricks. Firebricks are seldom laid dry; as a rule, they are laid with a thin layer of mortar between them. Firebricks are occasionally protected against heat and furnace atmosphere by a wash, which is either brushed on or sprayed (shot) by a furnace gun.

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* Very complete information on all details of brickwork can be found in *Bricklayer Trade Theory Manual,* United States Steel Corporation, Pittsburgh, Pa., 1949.

Other materials for walls and roofs of furnaces are plastic fireclay rammed into place, and hydraulic heat-resisting concrete (castable).

Further and more detailed information on these points is contained in later chapters.

chapter 2

The Heating of Solids in Industrial Furnaces

1. QUANTITY OF HEAT TO BE IMPARTED TO THE CHARGE

If solid pieces are to be heated in a furnace, heat must first be generated (liberated, released) in the furnace. The heat must then be transferred to the material which is to be heated (heating stock, charge, ware) and must finally be so distributed in the charge that the specifications of the metallurgical or ceramic engineer are met. These specifications cover final temperature of charge, uniformity of temperature in the charge, and time at temperature. Occasionally, rates of heating and of cooling are specified.

For a clear understanding of the heating process, it is advisable to begin with the physical properties of the material that is to be heated.

The heat that is to be imparted to the stock equals weight of stock times temperature-rise times mean specific heat of stock.

Handbooks contain information on the mean specific heat of metallic and non-metallic materials. The heat of important ferrous and non-ferrous metals may be obtained from the curves of Fig. 26. Figure 27 shows the heat content of iron and steel, illustrating the effect of varying percentages of carbon, as determined by several experimenters. The addition of the usual small amount of alloying elements, such as nickel, chromium, or manganese, changes the heat content of steel by only a negligible amount. The specific heat of "Inconel" (79.5 per cent nickel, 13 per cent chromium, 6.5 per cent iron) differs by only 1 per cent from the specific heat of mild steel.

The method of using the curves (Figs. 26 and 27) to find how much

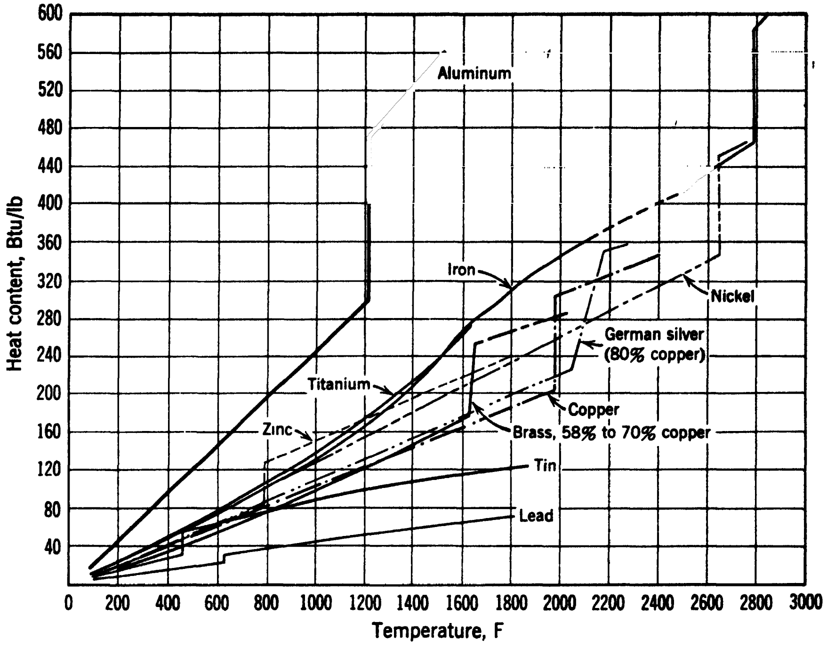

Fig. 26. Heat content of metals at various temperatures.

heat is absorbed by the stock, is illustrated by the following example: A bar of 0.30 per cent carbon steel, weighing 250 lb, has a temperature of 100 F when put into the furnace; it is to be heated to 2200 F. From Fig. 27, the heat content (above 0° F), when the bar is put in, equals 11 Btu per lb; and, when the bar is taken out of the furnace, if it is uniformly heated throughout to 2200 F, the heat content is 369 Btu per lb. Therefore, 250 lb times (369 — 11) (Btu per lb) = 89,500 Btu, the heat absorbed by the bar.

Extreme accuracy in the knowledge of heat content is seldom necessary, because the final temperature of the charge is uncertain within a range of ±10 F for low temperature (1400 F) and ±25 F for high temperature (2200 F).

If a furnace is to be used for heating materials in which the high temperature causes chemical reactions, specific heats and reaction heats should be looked up in handbooks of chemical engineering. An example is the burning of lime.

In the heating of ceramic materials, much heat is required for evaporating moisture.

HEAT REQUIRED FOR ENAMELING. In addition to imparting sensible heat, enameling requires heat for fusion and chemical reactions. For that reason it lies partly outside the scope of this book; but, since the metal to be enameled requires a large part of the total heat, data on enameling are furnished in the following paragraphs.

In vitreous-enameling practice, the batch composed of borax, quartz, feldspar, soda, cryolite, and metallic oxides is first melted to form a glass, which is then disintegrated by pouring into water, forming the "frit." The heat absorbed in the formation of the frit, including sensible heat in raising it to 2000 F, heat of fusion, and heat absorbed by chemical reactions, is 1540 Btu per lb of frit for the first or grip coat, or ground-coat enamel, and 1330 Btu per lb of frit for the cover-coat enamel (for representative batch mixtures).

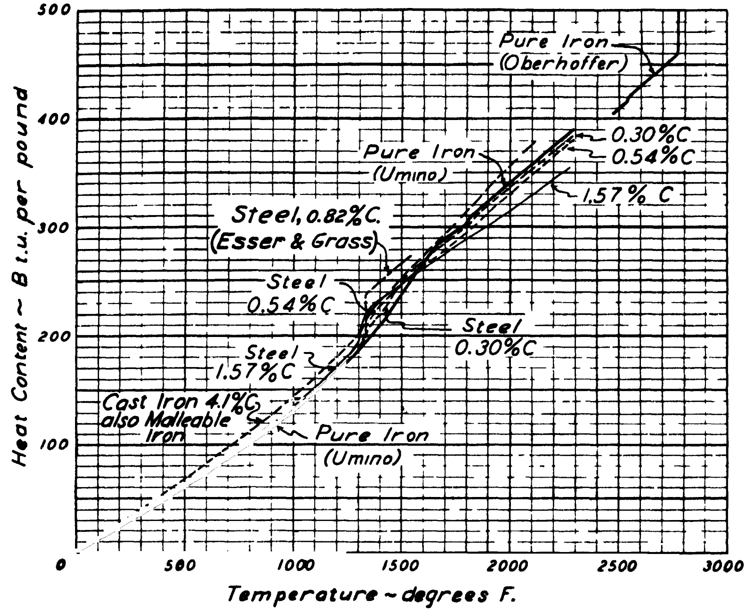

Fig. 27. Effect of varying percentages of carbon on the heat content of iron and steel.

The frit is ground to powder with the addition of about 12 per cent of its weight of clay and quartz or tin oxide, mixed with water (45 per cent by volume), coated on the metal to be enameled, and dried before it enters the enameling furnace. The heat absorbed by the enamel itself, when heated to 1650 F but not including drying, is 395 Btu per lb of grip-coat enamel, and 370 Btu per lb of cover-coat enamel. The weight of enamel applied is about 0.077 psf for the grip coat, and 0.108 psf for the cover coat, on each side of the metal.* The heat absorbed by the enamel, in heating to 1650 F, then is 61 Btu per sq ft for the grip coat, two sides, and 61 + 79 = 140 Btu per sq ft for the cover coat. The heat absorbed by the metal itself, if of 24 gage sheet steel (0.025 in. thick), is about 280 Btu per sq ft; hence, the heat taken up by the enamel is about 22 per cent as much as the heat in the metal during the first heating and 50 per cent during the second heating, amounts that are by no means negligible. The thicker the metal, the less are the percentages of heat absorbed by the enamel, and for castings, of course, they become very small. In addition, the supports which carry the ware through the furnace usually absorb as much heat as the metal plus coating together, although efforts have been made to reduce the weight of these fixtures by proper design.

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* Data from Ingram-Richardson Manufacturing Company, Beaver Falls, Pa., and Federal Enameling and Stamping Company, Pittsburgh, Pa.

JAPANNING. Japan coatings consist of a gum or resin, such as gilsonite; a small amount of pigment, such as lampblack; a drying oil, usually linseed oil; and thinning oils, chiefly benzene and kerosene, added for ease of application. The metal parts are usually coated by dipping in the liquid, and are then baked in an oven, at temperatures rising to, usually, 375 F, sometimes 450 F, for black japan, but seldom exceeding 200 F for colored finishes. Heat is absorbed in heating the pigment; and in fusing the resin. Some heat is evolved by the oxidation of part of the linseed oil. The japan film which remains, consisting of pigment, resin, and probably a small amount of oxidized oil, is about 27 per cent, by weight, of the original mixture, the film is about 0.0009 in. thick.

The heat absorbed is about 300 Btu per lb of the liquid mixture, or 1110 Btu per lb of the film, the density of the latter being estimated at about 68 pcf. Hence, for flat sheet steel, 20 gage (0.0375 in. thick), the heat absorbed by the coating, up to 450 F finishing temperature, is about 11.6 Btu per sq ft. This amounts to about 15 per cent of the heat in the steel itself, for each coat of japan.

ADDITIONAL HEAT REQUIREMENTS. In many heating operations, additional heat is needed. If the charge rests in containers or on trays, such supports must be heated. Water-cooled skids absorb heat. And if furnace and charge are to be heated together from cold conditions, the furnace walls may absorb almost as much heat as the charge. These items of heat are discussed in later chapters.

2. FLOW OF HEAT IN THE CHARGE

If a bar is heated by an electric current that flows through the bar, heat is generated uniformly in the cross-section of the bar. But in the great majority of heating processes heat enters the stock through its surface and flows from there into the interior of the stock or charge. Heat flow requires a difference of temperature in the stock. The effect of some of the variables in heat flow is illustrated by the simple case of steady flow of heat through a plate. The steady flow of heat is represented by the following equation:

$$\frac{\overset{\substack{\text{Heat flow,}\\ \text{Btu}}}{Q}}{\underset{\text{Hours}}{t}} \quad \overset{\substack{\text{Conductivity,}\\ \text{Btu per (ft, hr, F)}}}{C} \times \overset{\substack{\text{Area,}\\ \text{sq ft}}}{A} \times \frac{\overset{\substack{\text{Temperature}\\ \text{difference, F}}}{(T_1 - T_2)}}{\underset{\text{Thickness of plate, ft}}{s}} \qquad (1)$$

But in a charge that is being heated, the flow is seldom parallel (such as in a plate), and is never steady. It is transient, which means that the temperature at a given point in the stock changes as time goes on. In transient flow of heat, equation 1 is replaced by three differential equations, each of which holds for one of the three coordinates of space. The general solution of these differential equations, by means of which the temperature of a given point can be found for a given time, is usually impossible. The equations have been solved for a few simple shapes, such as the plate, the cylinder, and the sphere. The solutions have been incorporated in charts which are given in Appendix 1 (*a*), with examples for their use. These charts are approximations only, because they are based upon a conductivity that is constant, independent of temperature, and on constant coefficients of heat transfer, neither of which assumptions is correct.

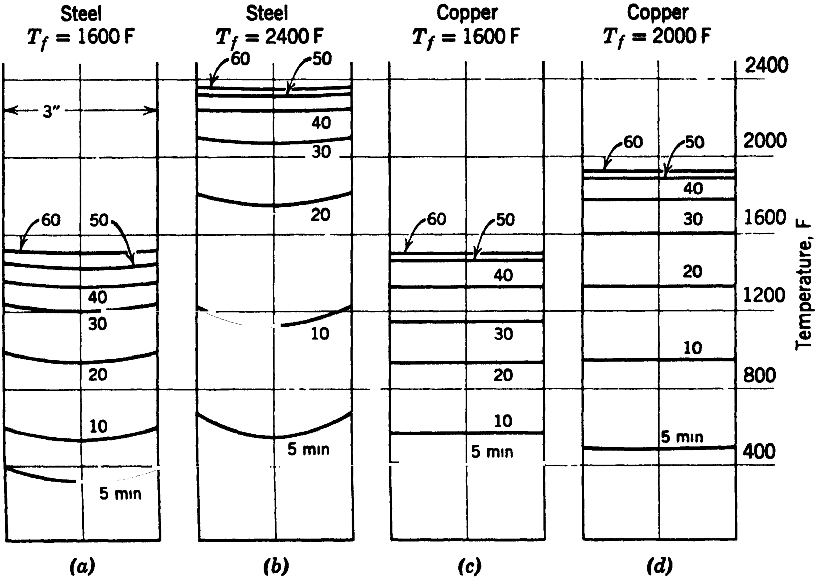


Fig. 28. Temperature rise in plates heated from both sides with different furnace temperatures.

In spite of these imperfections, the charts afford a good insight into the temperature distribution in solids that are being heated or cooled. Figure 28 was drawn from calculations that were made from Figs. 344 and 347, to show how the temperature rises in plates that are heated from both sides. Sketches *a* and *b* hold for steel, whereas *c* and *d* were drawn for copper. Sketches *b* and *d* were drawn for a furnace temperature that is much higher than the final metal temperature (high-speed heating). The numerals pointing to the curved temperature lines indicate the time in minutes that has elapsed from the beginning of the heating process.

Two conclusions can be drawn from these illustrations. The first is that, in high-speed heating, the temperature in steel is far from uniform, when the desired final temperature has been reached. The second conclusion is that, under comparative conditions, temperature is better equalized in copper than in steel, which fact of course, is the direct result of the difference in thermal conductivities.

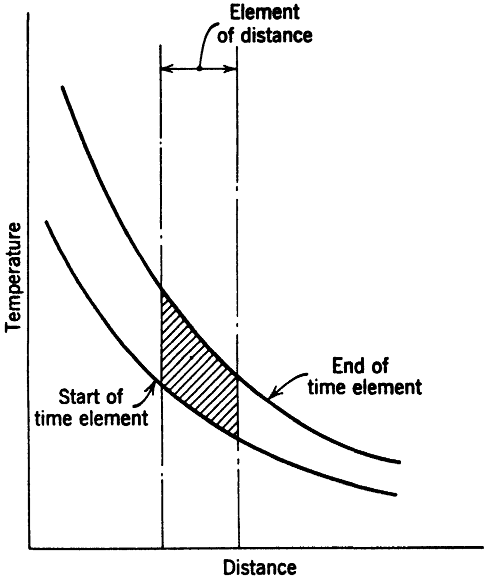


Fig. 29. Diagram showing rise in temperature of a strip or ring of material in an element of time.

Curves such as those of Fig. 28 can be obtained graphically by what is commonly known as the Schmidt method, which is explained in Appendix 1(*b*). In that method, finite small differences are substituted for the infinitely small differentials of the general equation. The Schmidt method has two advantages: Heating can start from any temperature distribution in the stock at the beginning of the heating period, whereas the use of the charts (Figs. 344–349) requires a uniform temperature at the start. The Schmidt method also can take care of variations in the coefficient of heat transfer.

Heat flow may also be investigated by drilling holes in a full-sized specimen and inserting thermocouples. The method is expensive, but it is often used.

From Fig. 28 it follows that thermal conductivity exerts a great influence upon the temperature distribution in solids in the flow of heat. However, thermal conductivity is not the only important property. Another property which is called diffusivity is also important. Diffusivity (sometimes called "temperature conductivity") equals

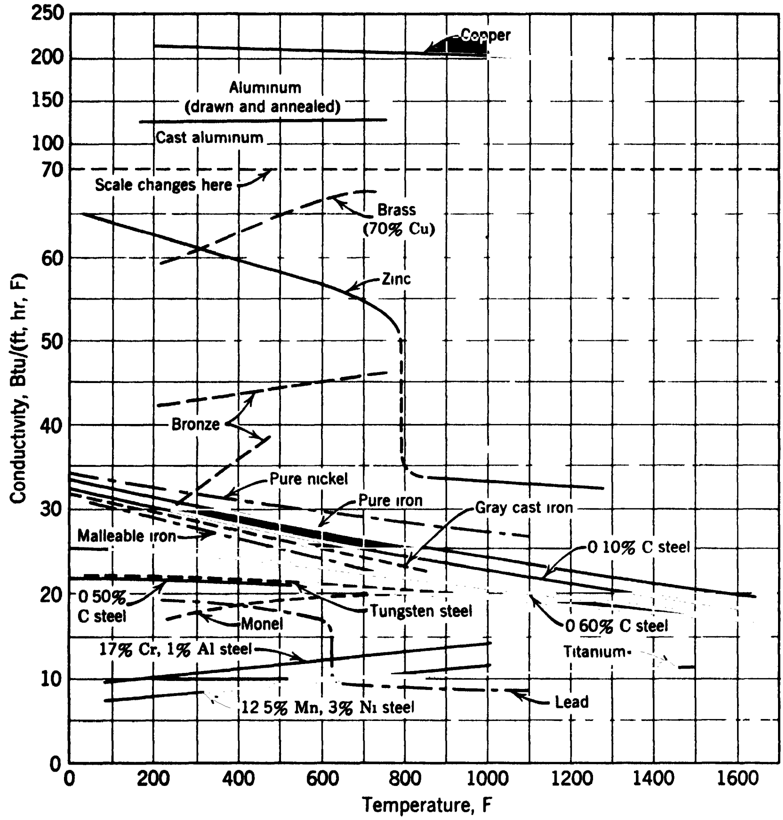


Fig. 30. Thermal conductivity of metals

$$\frac{\overset{\text{Btu per (ft, hr, F)}}{\text{Thermal conductivity}}}{\underset{\text{Btu/lb F}}{\text{Specific heat}} \times \underset{\text{lb/cu ft}}{\text{Density}}}$$

The effect of this quantity in transient flow of heat becomes clear from the following reasoning. In transient flow of heat, temperature-location curves are never straight, because straight lines indicate steady, constant flow of heat. In Fig. 29, the two curved lines represent temperature distribution in the solid at the beginning and at the end of a short interval of time. The cross-sectioned area indicates the rise of temperature in a strip or ring of the material being heated.

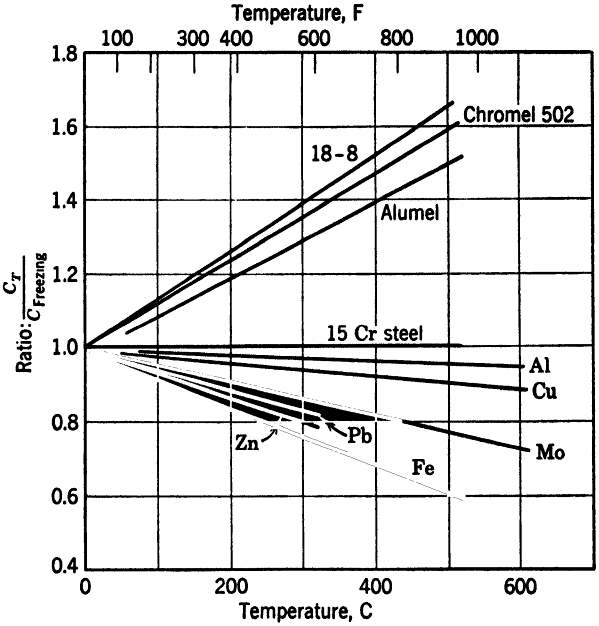


Fig. 31. Ratio of thermal conductivities at any temperature and at freezing temperature (32 F).

It also indicates the heat imparted to the element, which heat equals weight times specific heat times temperature rise. The greater the ratio

$$\frac{\text{Thermal conductivity}}{\text{Specific heat} \times \text{Density}}$$

the more rapidly the temperature diffuses (is conducted) into the region of lower temperature. The utility of "diffusivity" is shown in later chapters and in the Appendix.

Table 2 gives values of conductivity and of diffusivity of materials that interest furnace men, as far as they are known at the present time.

Thermal conductivity of solids varies with temperature, and so does diffusivity, because thermal conductivity is a factor of diffusivity. Specific heat and density vary but little with temperature, except in the case of steel, where the recalescence point introduces a discontinuity.

TABLE 2. Conductivity, Specific Heat, and Diffusivity of Metals at Ordinary Temperatures (100 F)

Compiled from Landolt-Börnstein Tables and Other Sources

Metal	Heat Conductivity, Btu per (ft, hr, F) at 100 F	Density, pcf	Thermal Capacity, Btu per (lb, F) at 100 F	Diffusivity or Temperature Conductivity at 100 F (Units are sq ft per hr)
Pure iron	33	490	0.110	0.61
Machinery steel	30	488	0.115	0.54
Tool steel	23	481	0.120	0.40
Cast iron (gray)	31	442	0.13	0.54
Malleable iron	(Same as cast iron)	458	0.122	0.55
Chrome steel, 3% Cr	21			
[(Varies with heat treatment)] 10% Cr	13	483	0.120	0.22
20% Cr	10			
Nickel steel, 5% Ni	18	492		
10% Ni	15			
30% Ni	5	500	0.119	0.09
Manganese steel, 10% Mn	7.2	498	0.125	0.12
Copper	220	558	0.104	3.8
Brass	58	530	0.092	1.2
Bronze	42	510	0.086	1.0
Manganese bronze	42			
Phosphor bronze	33	554	0.087	0.68
Monel metal	16	555	0.13	0.22
Nickel	33	537	0.103	0.60
Aluminum, cast	108	165	0.248	2.6
Aluminum, drawn and annealed	126	168	0.248	3.0
Aluminum alloy, 92% Al +8% Cu	88	180	....	(2.3)
Aluminum alloy, 8% Cu +1% other metals	75	178	0.214	2.0
Zinc	63	446	0.094	1.5
Die-casting metal, zinc base	54	432		
Lead, solid	19	708	0.031	0.87
Lead, molten	9.5	650	0.034	0.43

Figures in parentheses are estimated values.

The variation of thermal conductivity as a function of temperature is illustrated by Figs. 30 and 31. Figure 31 reveals a peculiar fact. The thermal conductivity of pure metals drops with rising temperature, whereas the thermal conductivity of alloys rises with temperature.

TABLE 3. Composition of Steels for Which Thermal Conductivities Are Given in Fig. 32

Composition in Per Cent

No.	C	Si	Mn	Cr	Ni	Mo	W
1	Pure Iron 99.95% Fe						
2	0.23	0.11	0.63	..	..	..	..
3	0.43	0.20	0.69	..	..	..	..
4	0.31	0.20	0.69	1.09	..	..	..
5	0.32	0.18	0.55	0.17	3.5	..	..
6	0.32	0.25	0.55	0.71	3.4	..	..
7	0.34	0.27	0.55	0.78	3.5	0.39	..
8	0.13	0.17	0.25	13.0	0.14	..	..
9	0.27	0.18	0.28	13.7	0.20	..	0.25
10	0.71	..	0.25	4.3	..	..	18.4
11	0.08	0.68	0.37	19.1	8.1	..	0.60
12	1.22	0.22	13.0	..	..	..	..
13	0.46	1.3	1.2	15.2	26.9	..	2.8

Steels of various compositions exhibit surprisingly great variations of thermal conductivity. This statement is proved by Fig. 32. At room temperature the thermal conductivity of steel No. 1 is six times as great as that of steel No. 13. This great variability makes accurate calculation of transient temperatures impossible.

The difference in thermal conductivity affects the flow of heat and the temperature distribution in steel, which fact is illustrated by Fig. 33 (*Industrial Heating,* June 1956, page 1224). The three curves were drawn for a skin temperature of about 1000 F. The curve in the middle was drawn for constant, average conductivity. Curve (1) was drawn for steel No. 1 of Fig. 32, whereas curve (13) was drawn for steel No. 13. The significance of these temperature distributions appears in the chapter that deals with the heating capacity of furnaces. Curves 1 and 13 are estimates because the differential equations of heat flow in metals with varying conductivity have not yet been integrated.

Solid material that is heated in industrial furnaces is not necessarily continuous. Very often the charge consists of separate individual pieces that are piled to various depths or of coiled material.

In charges of this description, heat flows in each solid piece as before, but flows from one piece to the adjacent piece through small contact

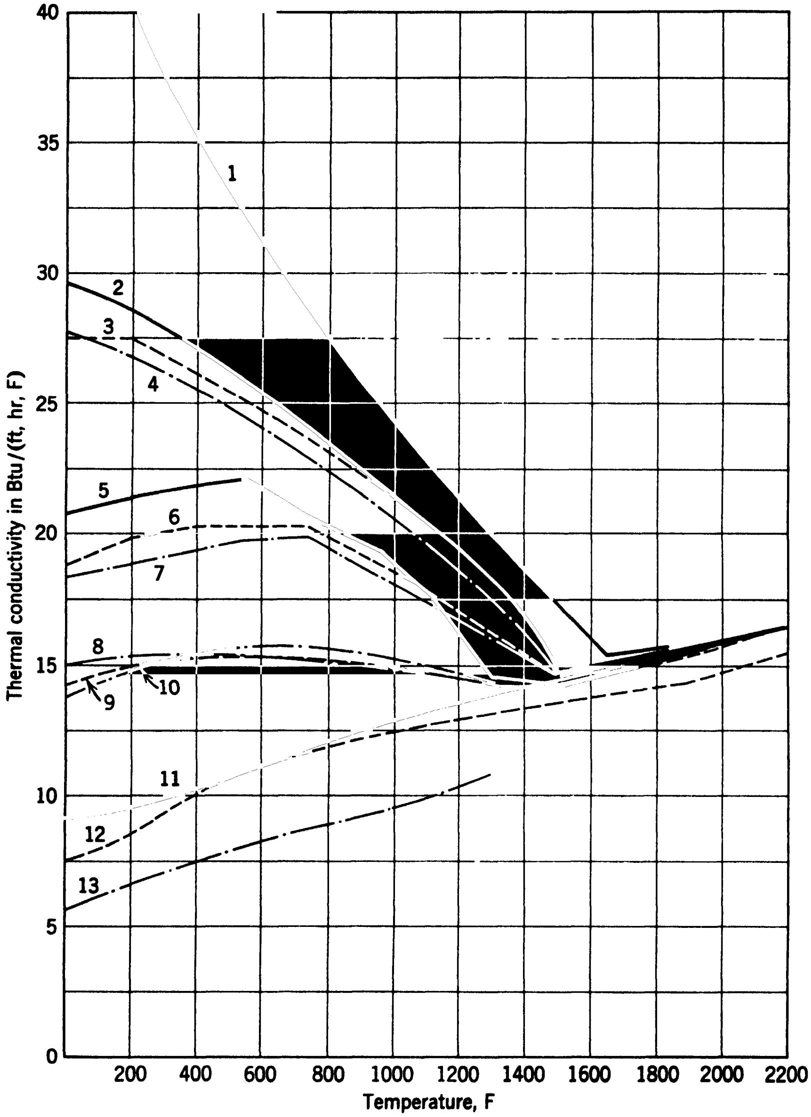
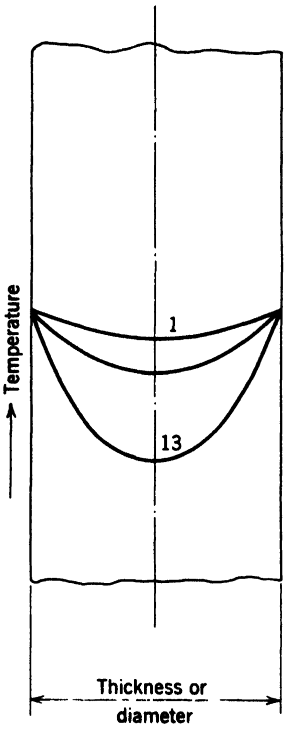


Fig. 32. Thermal conductivity of ferrous alloys given in Table 3.

surfaces and through gas-filled spaces, the thermal conductivity of which is very small. A pile of fish-hooks is a good example of low overall conductivity. Rapid heat flow in each piece of a piled charge is obtained by circulation of hot gases through the piled material. This situation leads to the discussion of the next step.

Fig. 33. Temperature distribution with same heating rate. Middle curve for average conductivity; others for steels in Fig. 32.



3. TRANSMISSION OF HEAT TO THE SURFACE OF THE CHARGE

In theoretical classification, heat is transferred by conduction or by convection or by radiation. In furnace practice, heat is transferred by a combination of all three means. Hundreds of papers and large books have been published on heat transfer.* In this book only those essentials of heat transfer are discussed that are helpful to designers and operators of industrial furnaces.

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* For a concise treatise on heat transfer, see Max Jakob, "A Survey of the Science of Heat Transmission," Purdue University, Industrial Bulletin 68. Elementary textbooks are Brown and Marco, *Introduction to Heat Transfer,* McGraw-Hill, 1942, and Jakob and Hawkins, *Elements of Heat Transfer,* John Wiley, 1957. More complete works are W. H. McAdams, *Heat Transmission,* McGraw-Hill, and Schack, Smith, and Partridge, *Industrial Heat Transfer,* John Wiley (out of print). The third (German) edition of the last book is very complete.

In heat transfer by conduction, the molecules of both the hot material and the cold material retain their relative positions, while the two bodies are in contact.

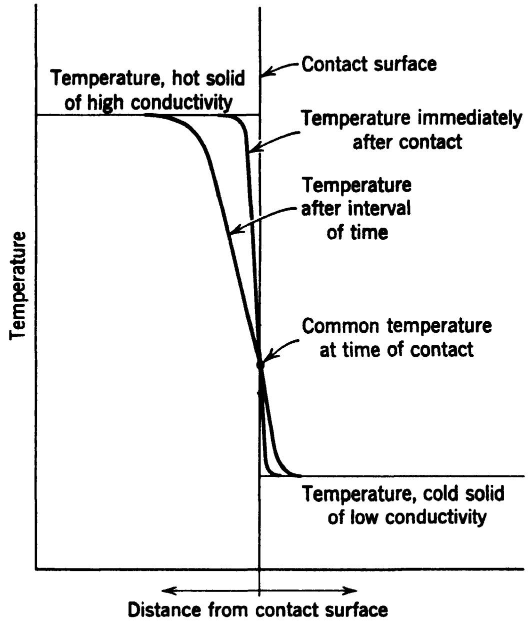

Temperature, hot solid of high conductivity
Contact surface
Temperature immediately after contact
Temperature after interval of time
Common temperature at time of contact
Temperature, cold solid of low conductivity
Temperature
Distance from contact surface

Fig. 34. Effect of conductivity and time on temperature gradients of two solids of different temperatures when heat is transferred by conduction.

(a) By Conduction. Heat transfer solely by conduction to the charge is rare in industrial furnaces. It occurs when cold metal is laid on a hot hearth. It also occurs, for a short time, when a piece of metal is submerged in a salt bath or a bath of molten metal. If two pieces of solid material are in contact (not separated by a layer of air or other fluid) the contacting surfaces instantly assume an identical temperature that lies somewhere between the temperatures of the contacting bodies. The temperature gradients at the contacting surfaces are inversely proportional to the conductivities of the contacting materials, as indicated in Fig. 34. The amount of heat transferred in unit time depends not only on the temperatures of the two bodies but also on the diffusivity and the shape of the contacting bodies.

In practice, comparatively little heat is transferred to (or abstracted from) the charge by conduction, although a frequent case of heat abstraction is the transfer of heat from a bloom to water-cooled skids.

Example. A bloom of 2000 F is pushed along a water-cooled 2½-in. skid pipe, to the top of which is welded a 1-in. square steel bar. This arrangement makes the distance from the interface to the water equal to 0.11 ft. The water has been softened and has 160 F. The contacting surfaces have a temperature somewhere between 2000 F and 160 F. Because of variations in thermal conductivity with temperature and because of difference in conducting areas on the hot and cold sides, the temperature of the interface is estimated to be 1000 F. The temperature difference then is $1000 - 160 = 840$ F. The average conductivity of steel between 1000 F and 160 F = 30 Btu per (ft, hr, F). The heat conducting area per foot of length of pipe equals 0.083 sq ft. Then the rate of heat transfer to the skid pipe equals $30 \times 0.83 \times 840/0.11 = 19{,}000$ Btu per (ft, hr).

When a piece of cold metal is thrust into molten salt or lead or other molten metal, the salt or lead freezes on the surface of the cold metal, and heat is transferred by conduction only. After a very short time, the solid jacket is melted. From that time on, heat is transferred by conduction and by convection. For that reason, discussion is postponed to the next section.

It should, however, be mentioned here that experimental determination of the heat transfer coefficient in the heating of metal in liquids is extremely difficult. For that reason, practice has adopted the method of recording the overall results. An example is shown in Fig. 75, which shows "time in bath" for good results as a function of thickness of strip and wire.

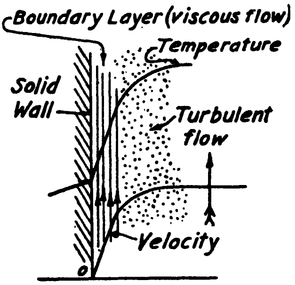


Fig. 35. Illustrating film theory of convection to surfaces.

(b) By Convection. By molecular and mass attraction, a layer of fluid is held at the surface of each solid. Heat is transferred through this stagnant layer by conduction. If a stream of gaseous fluid flows parallel to the surface of the solid, as indicated in Fig. 35, the vibrating molecules of the stationary and of the moving gases tangle. The skin of the stagnant layer is speeded up, and that of the moving stream is slowed down. The thickness of the stagnant (poorly conducting) boundary layer is reduced and heat transfer is increased.

Tests have shown that the transfer of heat per unit area in unit time is a function of mass velocity. For gaseous flow along plane walls, the equation is

$$\frac{Q}{A} = 7.28 \times q \times c^{0.78} \times (T_1 - T_2) \times t \qquad (2)$$

In words, heat transferred (Btu) to area A (sq ft) equals 7.28 times density q (pcf) of fluid times velocity c of fluid (ft per sec) to the 0.78 power times difference of temperature (F) between the fluid and wall times time t (hr).

Although use of the log-log slide rule has facilitated calculation with fractional exponents, most furnace engineers dislike them. The following equation avoids fractional exponents and is sufficiently close for all practical purposes:

$$\overset{\text{Btu}}{Q} = \overset{\text{sq ft}}{A} \times (1 + 2.7\ \overset{\text{lb/cu ft}}{q} \times \overset{\text{ft/sec}}{c}\ )\ \overset{\text{Difference of temp, F}}{(T_1 - T_2)} \times \overset{\text{hr}}{t} \qquad (3)$$

The use of the more correct equation is justified in the calculation of tubular heat exchangers, such as recuperators, but is out of place in the calculation of furnaces. Gas-velocities and gas temperatures vary from place to place in the furnace. The gas stream sometimes passes along the roof and does not contact the heating stock. In many furnaces, the charge consists of individual pieces that rest on trays. Such a charge has no plane surface, but rather an aggregate of cylindrical and spherical surfaces. Gases impinging upon them follow different laws of heat transfer. In such cases, equation 3 offers only a generalized idea of the influence of mass velocity upon heat transfer.

In furnaces that carry a temperature of 1100 F or lower, heat transfer by convection is of greatest importance. Velocities of circulating gases are purposely so high, that the coefficients equal 6 Btu (per sq ft, hr, F) and even higher. At low furnace temperatures (and in ovens) high coefficients, Btu per (sq ft, hr, F), can be obtained only by high velocities of furnace gases, because heat transfer by radiation at 1000 F is less than one-tenth of what it is at 2200 F. This situation is discussed in the following section.

(c) By Radiation between Solids. It is a common experience that hot solid bodies radiate heat. They radiate some heat even at low temperatures. The heat transferred by radiation from a hot solid to a colder solid equals the difference between the heat radiated from the hot body to the cold body and the heat radiated from the cold body to the hot body.

The heat radiated from a solid body is given by the equation:

$$\overset{\substack{\text{Btu per}\\ \text{(sq ft, hr)}}}{\frac{Q}{A}} = \text{Coefficient} \times \underset{\substack{\text{Absolute}\\ \text{temperature, F}}}{T^4} \tag{4}$$

The coefficient is exceedingly small and the fourth power of the absolute temperature is exceedingly great. For that reason, the radiation equation is commonly written in the following manner:

$$\frac{Q}{A} = \text{Coefficient}\left(\frac{T}{100}\right)^4 \tag{5}$$

The greatest possible value of the coefficient in equation 5 is 0.173. For any solid, the coefficient equals 0.173 times emissivity. The emissivities of materials commonly encountered in industrial furnaces are given in Fig. 36. Emissivity equals absorbivity. By this latter term is meant the heat which is absorbed by a body as a fraction of the greatest possible heat absorption in unit time and by unit area.

The coefficient 0.173 is called the blackbody coefficient. Equation 5 expresses the heat which a solid body emits by radiation; it does not express the heat transferred to the surroundings. The heat actually transferred is the difference between emitted heat and heat received by radiation from the surroundings. This difference (which equals the heat actually transferred by blackbody radiation) equals

$$\overset{\substack{\text{Btu}\\ \text{hr}}}{Q} = \overset{\substack{\text{Blackbody}\\ \text{coefficient}}}{0.173}\ \underset{\substack{\text{Area of}\\ \text{surrounded}\\ \text{body, sq ft}}}{A}\left[\overset{\substack{\text{Absolute temp.}\\ \text{of hotter body, F}}}{\left(\frac{T_1}{100}\right)} - \overset{\substack{\text{Absolute temp.}\\ \text{of colder body, F}}}{\left(\frac{T_2}{100}\right)^4}\right] \tag{6}$$

Neither the materials heated in furnaces nor the furnace walls are blackbodies. If both heat-exchanging bodies have the same emissivity, the coefficient 0.173 in equation 6 is multiplied by the emissivity. If the emissivities differ, the coefficient is composite, and the equation becomes, for two parallel plates,

$$\overset{\text{Btu}}{Q} = \frac{\overset{\substack{\text{Smaller area}\\ \text{sq ft}}}{A_1} \times \overset{\substack{\text{Time}\\ \text{hr}}}{t} \times 0.173}{\underset{\substack{\text{Emissivity}\\ \text{of } A_1}}{\frac{1}{e_1}} + \frac{A_1}{\underset{\substack{\text{Larger}\\ \text{area}}}{A_2}}\left(\underset{\substack{\text{Emissivity}\\ \text{of } A_2}}{\frac{1}{e_2}} - 1\right)}\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \tag{7}$$

Equation 7 is correct for radiation through vacuum or through transparent gases that do not absorb heat.

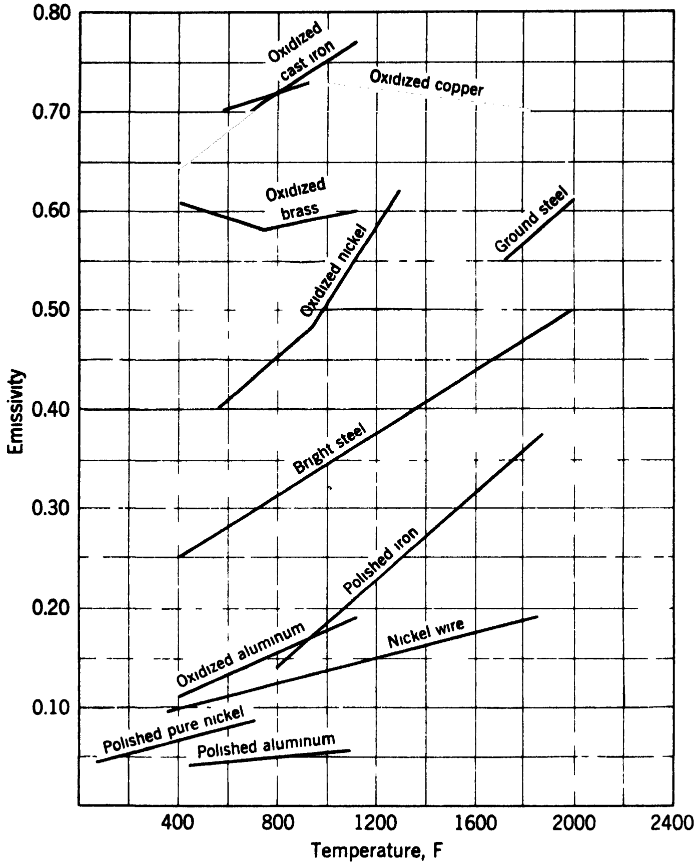


Fig. 36. Emissivity of metals.

The expression $T_1^4 - T_2^4$ can be factored

$$T_1^4 - T_2^4 = (T_1 - T_2)(T_1^3 + T_1^2T_2 + T_1T_2^2 + T_2^3)$$

If T_1 and T_2 are not far apart, the difference of the fourth powers of temperature approximately equals:

$$4(T_1 - T_2)\left(\frac{T_1 + T_2}{2}\right)^3$$

This expression teaches that even small temperature differences result in high rates of heat transfer at elevated temperatures. For instance, one degree F temperature difference at 2200 F causes about 5½ times as much transfer of heat as one degree temperature difference causes at 1000 F. It follows that the coefficient of heat transfer by solid radiation, Btu per (sq ft, hr, F) as used in the simple equation: $Q = kA(T_1 - T_2)t$, must vary widely with the temperatures of the heat exchanging solids. For blackbody radiation, this coefficient k equals $(T_1^3 + T_1^2T_2 + T_1T_2^2 + T_2^3)0.173/100^4$.

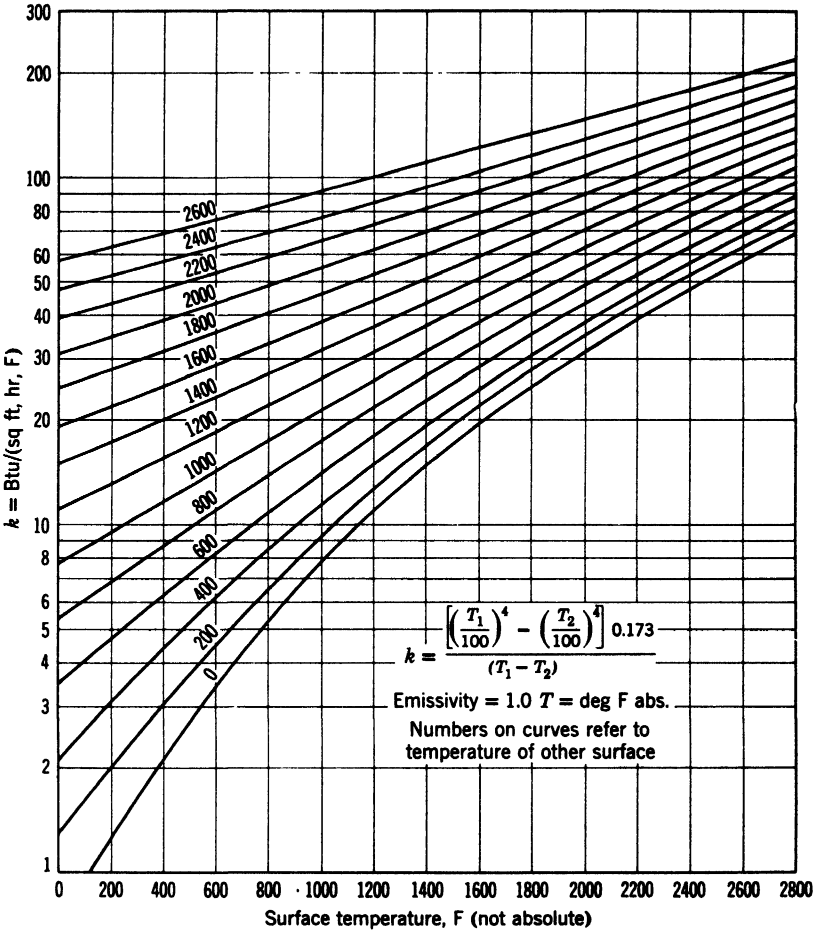


Fig. 37. Coefficient of heat transfer by radiation for emissivity = 1.0.

The extent to which the heat transfer coefficient varies is readily seen from the nest of curves in Fig. 37, where the coefficient appears as ordinate, while the heat exchanging temperatures appear as abscissae and parameters. The heat transfer coefficients in Fig. 37 are correct for blackbody radiation. They must be modified for bodies the emissivities of which are smaller than 1.0.

Example for Heat Transfer by Radiation between Solids. A single 4-in. by 4-in. steel billet, which rests on ledges, is being heated in a large electrically heated furnace that has a wall temperature of 1800 F. At the time when the billet has reached a surface temperature of 1200 F, what is the rate of heat input per foot of length? The heat-absorbing surface per foot of length equals 4 × 4/12 = 1.33 sq ft. The emissivities of wall and of oxidized steel are alike and equal 0.9. The problem can be solved by using either equation 6 or else Fig. 37. From this latter illustration, the coefficient of heat transfer by blackbody radiation is found to be 53 Btu per (sq ft, hr, F). The heat transferred by radiation equals 53 × 600 × 0.9 × 1.33 = 38000 Btu per (sq ft, hr). This heat absorption raises the temperature of the billet about 5° F per minute.

If the billet were made of copper, which has an emissivity of 0.5, equation 7 would have to be used. If emissivities of wall and of stock are not alike, the ratio of wall surface to stock surface enters into the calculation. If that ratio be *four*, the factor by which the figure gained from Fig. 37 must be mutiplied equals

$$\frac{1}{1/0.5 + 1/4\,(1/0.9 - 1)} = 0.48 \quad \text{(see equation 7)}$$

and the rate of heat transfer by radiation equals 53 × 600 × 0.48 × 1.33 = 20,300 Btu per (hr, ft of billet length).

The above given law of radiation, equation 6, is correct for heat exchange between a small ball and a surrounding large shell. For all other conditions, the equation must be modified as explained in later chapters.

Equations 6 and 7 have limitations. They apply only in those cases where radiation passes either through empty space or through gases that do not absorb heat.

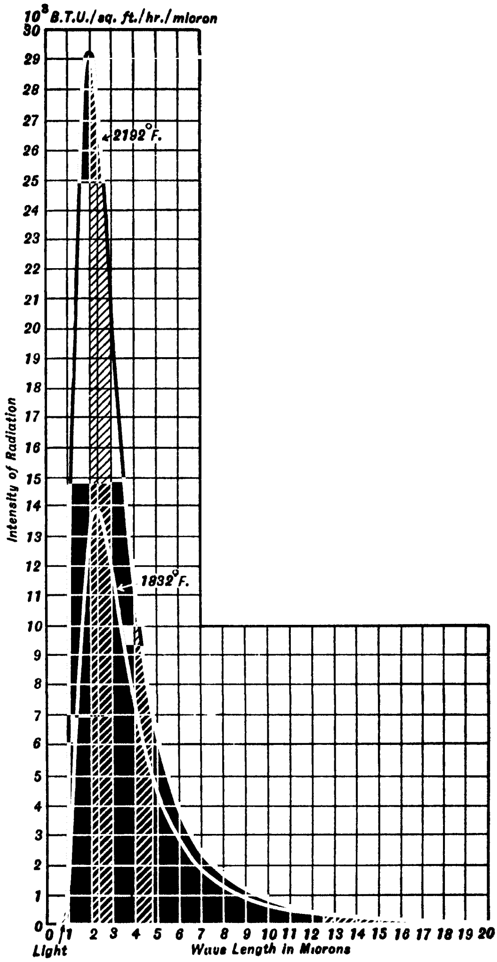
The fact that some furnace gases radiate and absorb heat leads to the next section.

(d) Radiation between Clear Gases and Solids. In radiation from products of combustion to solids, a difference must be made between radiation from clear gases and radiation from so-called luminous flames. Radiation from clear gases does not follow the fourth-power law (except by the artifice of using a variable emissivity), nor does it follow any other exponential law* as may be seen from the following explanation. The only clear gases that radiate appreciably are those having three or more atoms per molecule, such as CO_2, H_2O, SO_2; carbon monoxide, although diatomic, also gives off some radiation. The other diatomic gases, such as O_2, N_2 (and their mixture, air), and H_2 have only negligible radiating power. The reason why

* Attempts have been made to find equations for gaseous radiation. The equations that have been developed are much too complicated for practical use.

gaseous radiation does not follow the fourth-power law is this: Gases do not radiate in all wavelengths, as do solids; instead, each gas radiates only in three or four rather sharply defined bands or channels of wavelengths. In Fig. 38 the heavy lines show the variation of intensity of radiation with wavelength (Planck's Law) for two temperatures. The whole area under each curve represents blackbody radiation from solid surfaces; the section-lined areas represent the radiating bands for carbon dioxide.

Fig. 38. Intensity of radiation as a function of wavelength, and selective radiation of carbon dioxide gas.



Radiation from clear gases depends not only on their temperature, but also on their composition (percentage of each triatomic gas present), and on the thickness of the gas layer, as well as (to some extent) on the shape of the gas layer. Calculation of heat transfer from radiating clear gases to solids and vice versa is, up to the present time, based solely on observation (tests). The results of the tests were laid down by the experimenters either in tabular form or else

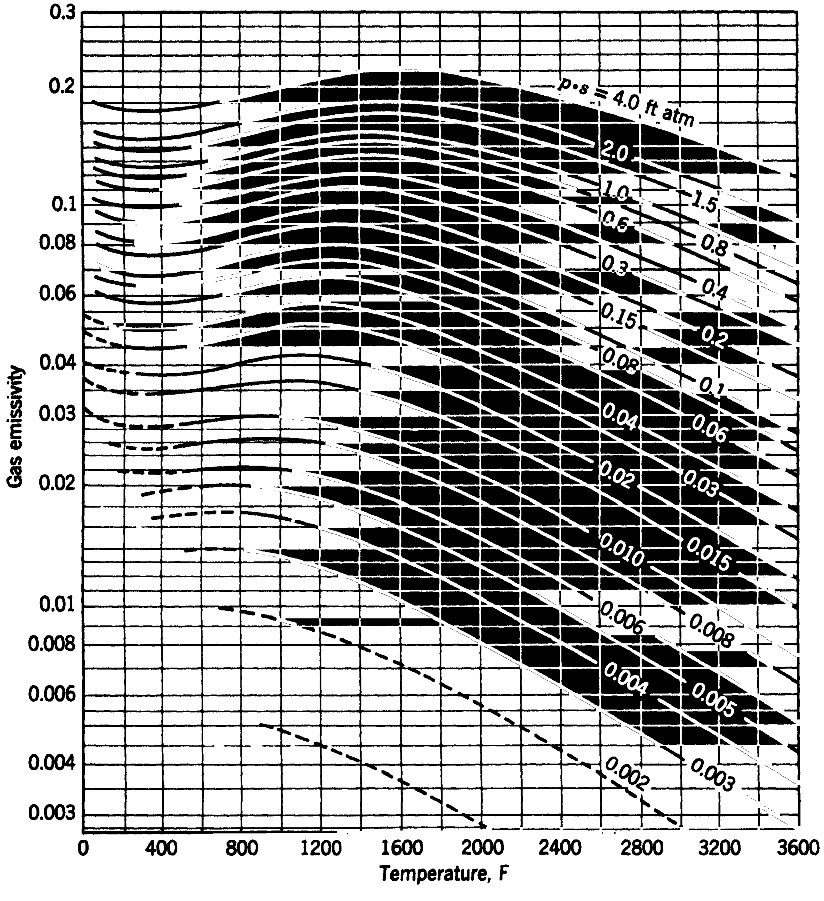


Fig. 39. Emissivity of carbon dioxide for various values of $p \cdot s$ (partial pressure times thickness of gas layer).

in graphs. The best known of these graphs furnishes the emissivity (absorptivity) of gases as a function of temperature, partial pressure, and thickness of gas layer. Charts for emissivities of CO_2 and of H_2O are shown in Figs. 39 and 40 (taken with permission from *Heat Transmission*, McAdams, Figs. 28 and 29). As previously indicated, the fourth-power law of radiation may be applied for calculation of gas-radiation by making use of Figs. 39 and 40.

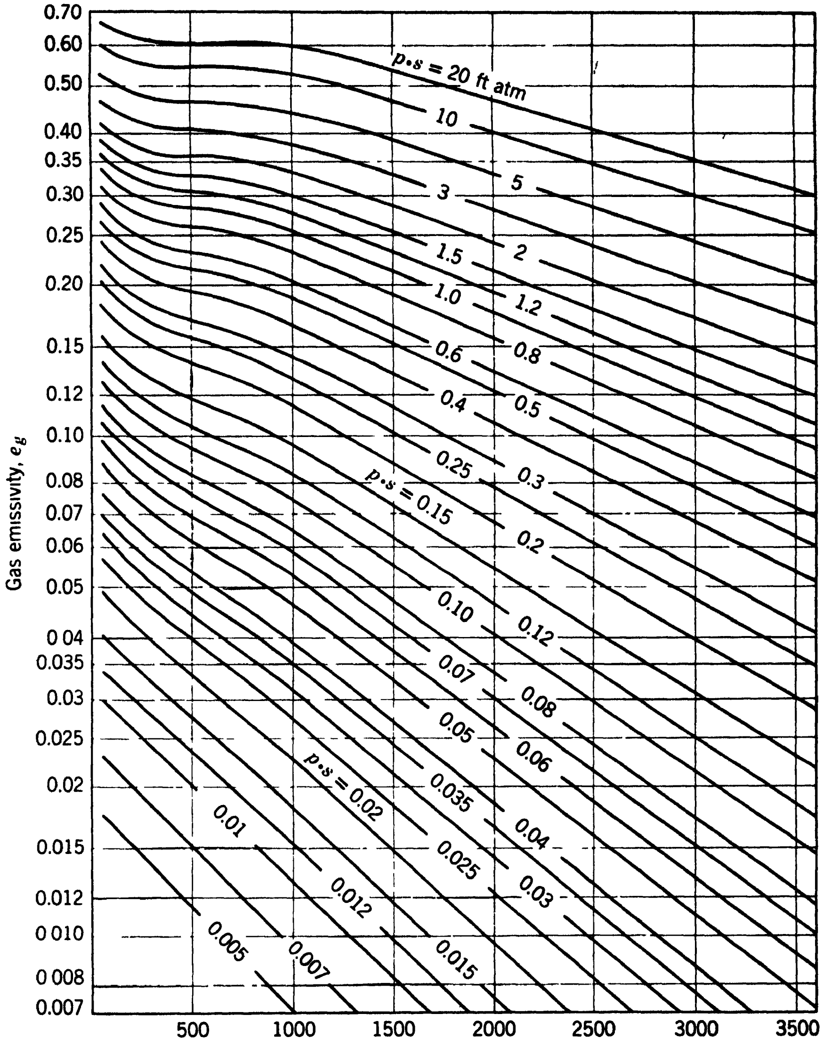


Fig. 40. Emissivity of water vapor for various values of $p \cdot s$.

A simpler method* which suffices for all practical purposes is illustrated by Figs. 41 and 42.

In these illustrations, the abscissae represent the arithmetical means between gas temperatures and temperatures of walls or stock. The ordinates are heat-transfer coefficients which are to be multiplied by the temperature difference between gas and surface of solid in order to obtain the heat transfer per square foot and hour, on the basis of 100 per cent emissivity of wall or stock. The inscriptions on the parameter curves refer to the product $p \times s$, which equals partial pressure of gas (atmospheres) times thickness of gas layer (ft). For superheated water vapor, the heat-transfer coefficient is not directly a function of $p \times s$, but is subject to a correction, which however, is so small for furnace conditions that it can safely be disregarded by furnace-engineers.

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* Schwisdessen, "Die Strahlung von Kohlensäure und Wasserdampf," *Archiv für des Eisenhüttenwesen,* October, 1940.

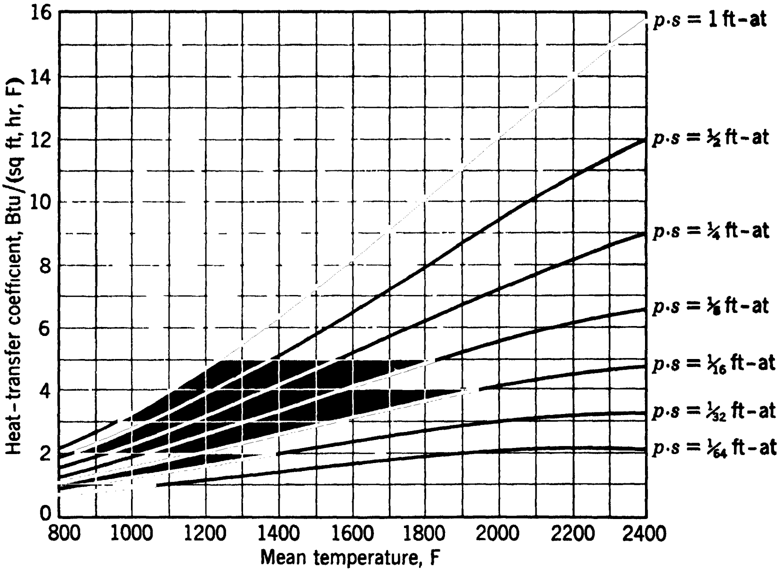

Fig. 41. Heat-transfer coefficients for radiation from carbon dioxide to a solid.

Example. Products of combustion of methane (CH_4) with neither excess nor deficiency of air contain, by volume, 9.5 per cent CO_2, 19 per cent H_2O, and 71.5 per cent N_2. Hence, the partial pressure of CO_2 is 0.095 atm and the partial pressure of H_2O is 0.19 atm. Let the depth or thickness of the gas layer be 3.17 ft. Then the product $p \times s$ equals $0.095 \times 3.17 = 0.3$ atm ft for CO_2 and $0.19 \times 3.17 = 0.61$ atm ft for H_2O. Let the temperature of the products of combustion be 2300 F, and that of the wall or stock 2000 F. Then the arithmetical mean temperature is $\frac{1}{2}(2300 + 2000) = 2150$ F. With reference to Fig. 41 (CO_2), vertically above 2150 F from the intersection with the $ps = 0.3$ curve (interpolated), a horizontal line leads to the desired heat-transfer coefficient at the left. In this example, the coefficient equals 8.3. In the same manner the coefficient for water vapor is found to be 15, from Fig. 42. No correction need be made for the peculiar behavior of water vapor as long as the mean temperature is above 1200 F. The total heat-transfer coefficient is 8.3 + 15 = 23 Btu per (sq ft, hr, F) if the two radiations are added directly. Five per cent (this is an average value) should de deducted from this figure because the radiating bands of CO_2 and H_2O partly overlap in the spectrum. A further reduction of the coefficient is necessary because the absorptivity (which equals emissivity) of the walls or of the charge is not 100 per cent. With an average value for absorptivity of 0.9, the heat-transfer coefficient

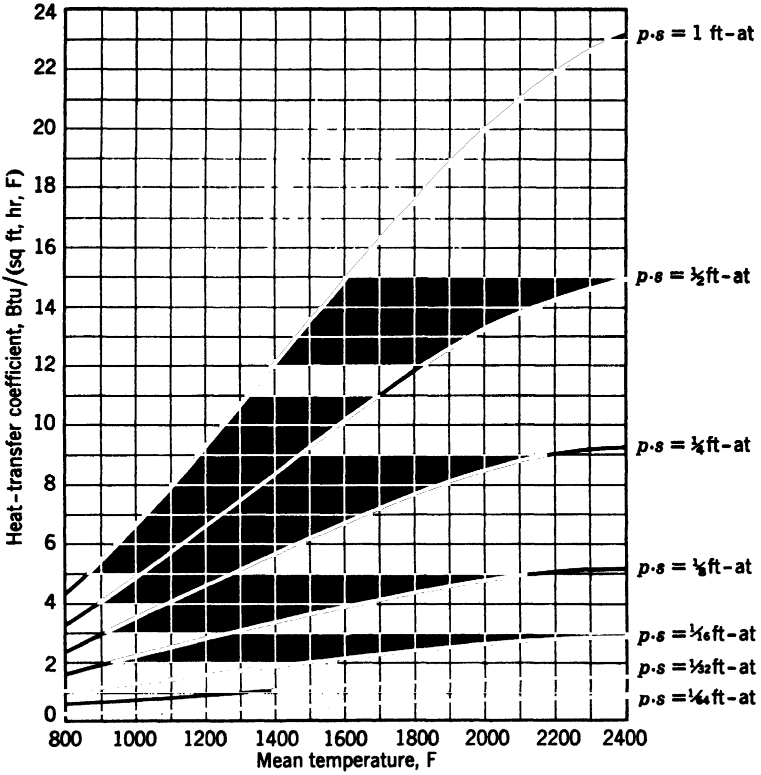

Fig. 42. Heat-transfer coefficients for radiation from water vapor to a solid.

becomes $0.9 \times 0.95 \times 23.3 \cong 19.8$. The heat transmitted to each square foot in 1 hr then is 300 (which is the temperature difference) times 19.8, or 5940 Btu.

It may be judged that the curves of Figs. 41 and 42 are too far apart for accurate interpolation. However, greater accuracy would be an illusion, because the temperature of the products of combustion varies from point to point, horizontally as well as vertically. Moreover it is very difficult to measure the temperature of products of combustion.

For rough and quick estimates of radiation from clear gases, Fig. 43 is useful. The range over which it may be applied is indicated in the legend.

The heat transfer coefficient derived from Figs. 41 and 42 are correct for a gas semisphere that radiates to a point in the center of the base. In this case the length of the "radiating beam" is constant. For gas volumes of other shapes equivalent "beams" have been calculated by tedious graphical and analytical methods. The most important values are given in Table 4.*

In contrast to the laws of heat transfer by convection, the laws of gaseous radiation contain no velocity factor; and yet velocity of the gases is of importance because, if a stationary hot gas radiates to colder surfaces, the gas necessarily loses temperature and finally becomes just as cold as the surrounding surfaces. To maintain active radiation, the radiating gas must continually be replaced by fresh hot gas. A gas that radiates to a cold surface becomes colder and colder in the direction of the gas travel. The higher the velocity of the gas, the lower is the drop of temperature of the radiating gas

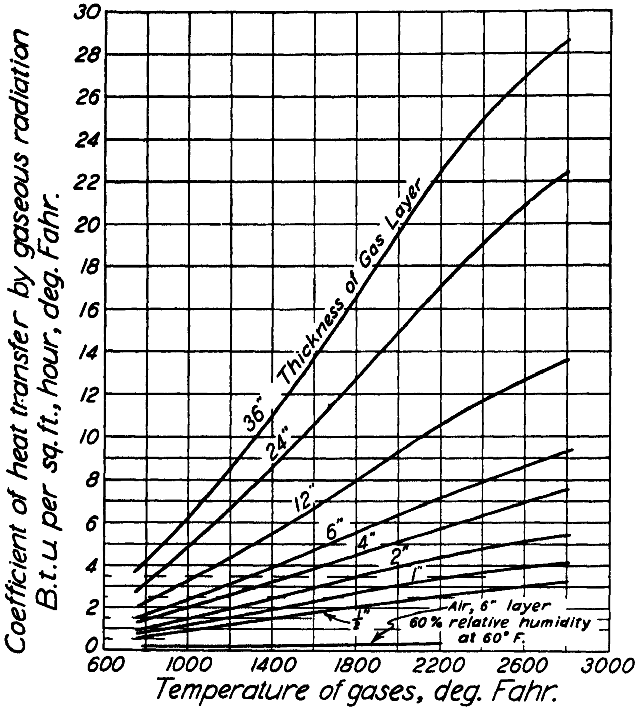

Fig. 43. Heat transfer by radiation from gases containing 12% CO_2 and 12% H_2O by volume. Values are decreased somewhat if temperature difference exceeds 350 F and are not to be used for temperature differences exceeding 500 F. Multiply values read from curves by radiation coefficient or emissivity of receiving surface: 0.91 for clean brick or tile or rough iron surfaces, 0.70 for glazed brick surfaces.

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* H. C. Hottel and R. B. Egbert, "The Radiation of Furnace Gases," in a paper read before the American Society of Mechanical Engineers in December, 1940. At the end of the paper a very complete bibliography is given on radiation of

along its travel. The temperature drop of the gas can be computed by means of a step-by-step method. (See Chapter 4, "The Heating Capacity of Continuous Furnaces.")

(e) Radiation from Luminous Flames. It is a common experience that heat transfer from a flame is greater than that from a clear gas having the same temperature. The difference in rate of heat transfer is quite noticeable in furnaces for reheating steel and even more so in furnaces for melting glass and for melting steel (open-hearth furnaces). The difference becomes more pronounced as the temperature goes up because the higher the temperature, the greater is the radiating power of each gas molecule, but (and this is important) the smaller is the number of radiating molecules in unit volume. Furthermore, the bands of gaseous radiation (see Fig. 38) retain their position, independent of temperature, whereas the point of greatest intensity of solid radiation moves toward shorter wavelengths (away from the bands) at higher temperatures. From these facts it follows that, with increasing temperature, radiation from clear gases cannot grow as rapidly as radiation from solids.

TABLE 4. Average Length of Radiation Beam, s

Shape of Gas Volume	Radiating Beams
Sphere	0.6 diameter
Infinitely long cylinder	0.9 diameter
Right circular cylinder, height = diameter, radiating to whole surface	0.9 diameter
Same, radiating to spot on center of base	0.6 diameter
Space between infinite parallel planes	1.8 distance between planes
Cube	0.6 edge
1 by 2 by 6 rectangular parallelepiped radiating to any of the faces	1.06 shortest edge

The application of these beams in the calculation of furnace capacity is explained in Chapter 3.

The word flame is often, but incorrectly applied to very hot, clear and almost invisible gases. Flames are always luminous. If a flame is cooled so as to be no longer luminous, it becomes smoke. The luminosity of flames is caused by glowing smoke particles. In flames that result from the combustion of solid fuels, glowing ash particles contribute to the luminescence. Depending upon number and size of the solid particles, the emissivity of a flame lies between the emissivity of clear gases and a high value of 0.95, as measured by Trinks and Keller.* This high emissivity exists in a short section only of the flame travel as shown in Fig. 44 which represents a typical relation between distance from burner and emissivity of flame. After leaving the burner, the fuel-air mixture requires time for reaching a temperature at which the hydrocarbons are cracked and for bringing the resulting smoke particles up to the temperature of the burning gases. As the flame travels along, formation of new smoke particles and combustion of previously formed particles balance. A short distance beyond that point no new smoke particles are formed, the smoke particles are gradually burned, and finally, a point is reached where the flame is extinguished (all smoke particles have been burned). Many furnace engineers like to have the flame extinguished when the products of combustion pass into the vent or flue. If the flame is still very luminous at that point, smoke appears at the top of the stack.

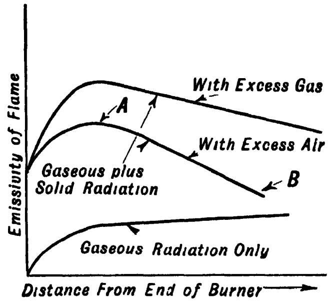

Fig. 44. Variation of emissivity of luminous flame with distance from burner.

Flame radiation is a function of many variables, the most important of which are: Composition of fuel, fuel to air ratio, temperatures of fuel and of air, rate of mixing of fuel and of air, thickness of flame, distance from burner. The composition of the fuel has a very great effect on flame luminosity. It is difficult to burn tar without flame and it is equally difficult to produce a flame when blast furnace gas is burned.

The effect of the fuel to air ratio upon emissivity is equally great. Figure 44 shows that, with excess natural gas, the emissivity is considerably greater than it is with excess air. It also shows that the luminosity drops off quickly, if an excess of air is provided. The illustration also shows to what extent flame radiation consists of gaseous radiation and radiation from solid particles.

The effect of mixing fuel with air upon luminosity and the means for adjusting the mixture are discussed in Volume II.

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* In 1956 R. Sherman presented, before the American Society of Mechanical Engineers, a paper on flame radiation, giving results obtained by various experimenters on flame radiation, expressed in Btu/(sq ft, hr).

In the Sherman collection of data on flame radiation referred to above, peak values of 200,000 Btu per (sq ft, hr) for flames from tar pitch or from residual oil are given. When referred to the whole flame length, the radiation is much lower. In Fig. 45 (Sherman page 10, Fig. 14) flame emissivity is given as a function of C/H ratio in the fuel. The average emissivity (for the whole flame length) may, for practical purposes, be taken as one half of the values given in Fig. 45.

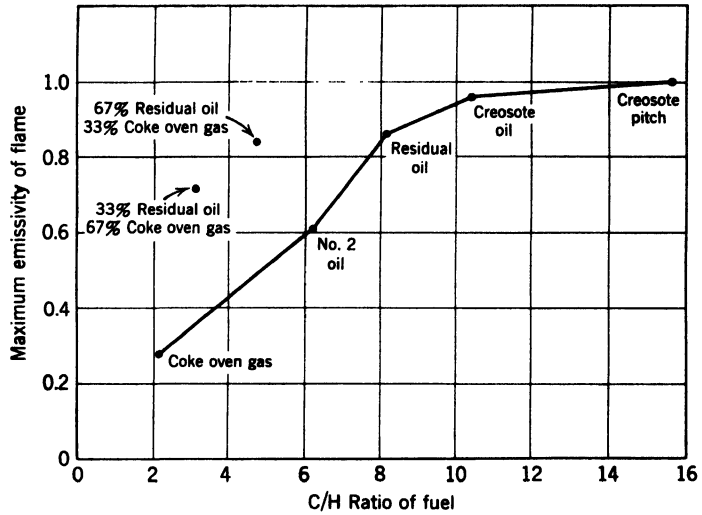

Fig. 45. Flame emissivity as a function of C/H ratio in fuel (Sherman).

A flame that has a high emissivity radiates almost like a solid body. With such a flame the furnace roof assumes flame temperature, if the flame is either wide or else travels close to the roof.

Recognition of the fact that products of combustion radiate not only to the heating stock but also to the walls leads to the next section.

4. THERMAL INTERACTION IN FURNACES

The flow of heat in a furnace is diagrammatically illustrated in Fig. 46 which represents a furnace that is heated by products of combustion.

Surface elements of the walls and of the charge, such as (3) and (6) are convection-heated by hot gases flowing along the walls and the charge (heating-stock). A molecule of the gases, such as indicated by (9) radiates in all directions, for instance to (2) and (8). These surface elements, in turn, radiate in all possible directions, for instance to (1), (5), and (7) from (3), and to (4) from (7), etc.

From Fig. 46 it follows that each particle of the products of combustion as well as each element of the surfaces of walls and charge radiate in all possible directions. This fact makes accurate calculation of heat transfer in a fuel-fired furnace extremely difficult, if not impossible. Proof for the correctness of this statement is furnished by the very large number of variables, including: temperature, composition, and velocity of the products of combustion, thickness of layer of gases between walls and charge, ratio of wall surface to exposed surface of charge, emissivities of walls and of charge.

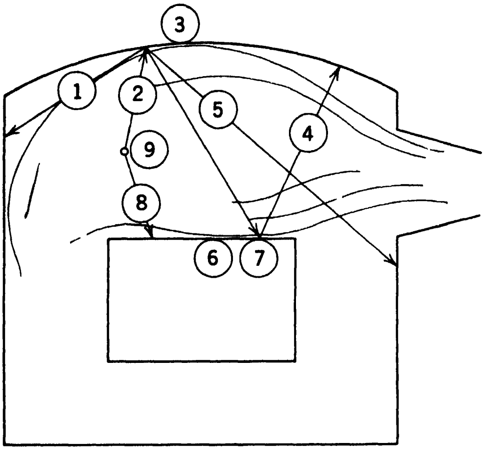

Fig. 46. Diagrammatic representation of heat transfer within a furnace enclosure.

Many furnace engineers do not fully understand these relations. For that reason, it appears advisable to compute a simple case and to proceed to more difficult cases from there on. The calculation of heat-transfer coefficients for the simple case was made for products of combustion of average natural gas (no excess air, no excess gas) that flow along the heating stock and the walls with such a velocity that the coefficient of heat transfer by convection equals 3 Btu per (sq ft, hr, F). The furnace is of a simple type. A cylindrical solid piece is suspended concentrically in a cylindrical furnace. The calculation was made for five temperatures of products of combustion. For each such temperature, a number of temperatures of heating stock were selected. For each of these combinations, different thicknesses of gaseous layers were selected, and finally, for each of these combinations of three variables, a different ratio of surface of stock to surface of walls was selected. For each of these combinations of variables the probable wall temperature was estimated, and a new wall temperature was calculated on the basis of the estimated temperature and of gaseous radiation. If estimated and calculated wall temperatures coincided, the overall heat transfer coefficient was correct. If they did not coincide, the calculation was repeated with a new estimate of wall temperature. A sample calculation is given in Appendix II. The calculations were made for emissivities of walls and stock equal to 0.9. The results of the calculations were laid down in Tables 5 and 6. Table 5 contains coefficients of heat transfer (from products of combustion to heating stock) that apply in the various conditions. It is readily observed that (for any one temperature of products of combustion) the coefficient varies not only with the temperature of the stock but also with the ratio of surfaces of stock and wall, and, in addition, with the thickness of the layer of products of combustion. Table 6 contains temperatures of furnace walls for the same conditions that were selected for Table 5. The differences between temperatures of furnace gases and wall are considerable.

EXAMPLE. A steel cylinder of 24 in. diameter is being heated concentrically in a cylindrical furnace having a diameter of 4 ft. Then ratio of surfaces (stock to furnace) equals 0.5, and s = thickness of gaseous layer equals 1 ft. The gases are products of combustion of natural gas; they have a temperature of 2400 F. At a time when the surface of the stock has reached a temperature of 1200 F, what is the rate at which heat is transferred to the stock, and what is the wall temperature?

From Table 6, the wall temperature is read to be 1740 F. From Table 5, the coefficient of heat transfer is read to be 25.6 Btu per (sq ft, hr, F). The heat transfer to each square foot of surface of stock $= 25.6(2400 - 1200) =$ 30,720 Btu per hr. Per foot of length, the rate of heat transfer equals $6.28 \times 30,720 = 193,000$ Btu per hr.

Other tables on the order of Tables 5 and 6 can be calculated for other fuels. The calculations are not difficult, but are very long.

As previously mentioned, Tables 5 and 6 were calculated for the theoretically correct ratio of air to natural gas. With excess air, gas radiation is smaller; the effect is the same as that caused by a thinner layer of furnace gases. Both, coefficient of heat transfer and wall temperature drop. With some excess gas (rich mixture) the products of combustion are still clear, but intermediate products of combustion are formed, which radiate. Coefficient of heat transfer and wall temperature rise. As luminosity (opacity) of flame is increased, while flame temperature is kept constant, both coefficient of heat transfer and wall temperature rise. When the flame becomes so luminous as

TABLE 5. Heat Transfer Coefficients

Gas Temp. F	Stock Temp. F	$s = 1$ ft Ratio R				$s = 2$ ft				$s = 3$ ft				$s = 4$ ft			
		0.1	0.3	0.5	0.7	0.1	0.3	0.5	0.7	0.1	0.3	0.5	0.7	0.1	0.3	0.5	0.7
2800	200	30.6	24.7	18.6	17.0	38.0	30.6	24.7	21.2	41.8	34.2	28.6	25.5	46.4	36.8	31.3	27.5
	400	36.1	26.7	21.2	18.3	44.8	33.2	26.6	23.2	49.4	37.7	31.1	27.8	54.7	40.8	34.0	29.8
	800	42.8	31.1	24.2	21.1	53.0	38.5	31.1	27.1	58.3	43.8	36.0	32.0	64.6	47.6	39.2	34.5
	1200	50.5	36.5	29.0	24.6	62.5	45.3	36.8	32.0	68.9	54.1	43.0	37.8	76.3	57.0	46.8	41.0
	1600	59.6	43.0	33.5	28.5	73.7	53.8	43.2	37.3	81.3	61.5	50.5	44.4	90.2	67.2	55.1	48.5
	2000	70.4	50.5	39.8	34.0	87.0	64.7	51.0	45.2	96.0	74.0	60.9	52.7	106.6	80.0	66.7	58.5
	2400	83.0	59.8	46.0	39.6	102.5	75.5	61.5	53.0	113.2	86.9	72.0	62.0	126.0	96.0	79.5	70.0
2400	200	27.7	21.2	16.7	15.0	34.5	25.5	21.3	18.3	37.6	27.9	24.0	21.9	38.8	28.7	25.8	23.0
	400	31.2	23.1	18.1	16.3	37.4	27.7	23.3	20.3	42.3	31.3	26.5	23.9	43.6	32.3	28.4	25.6
	800	36.9	27.3	21.5	19.1	44.6	33.0	27.4	24.2	50.6	37.5	31.6	28.0	54.0	40.0	33.6	30.2
	1200	44.6	33.0	25.6	23.0	55.2	40.8	33.3	29.2	60.8	45.5	38.5	34.2	64.8	48.2	41.0	37.0
	1600	54.6	40.4	31.8	27.6	65.4	48.3	39.3	34.2	75.5	56.2	46.9	41.6	79.3	58.7	49.5	44.2
	2000	67.6	50.1	39.2	32.8	79.8	59.6	49.0	42.6	92.5	68.5	57.7	51.0	97.0	71.8	60.2	56.2
2000	200	20.8	17.3	14.2	12.8	22.5	18.5	15.5	13.4	24.8	21.4	19.0	17.2	27.5	24.4	22.2	21.4
	400	23.3	18.4	16.0	14.3	24.5	20.4	17.3	15.2	27.8	24.0	21.0	19.2	31.3	27.8	24.9	24.1
	800	28.2	23.5	19.4	17.3	29.8	24.8	20.6	18.6	34.2	29.6	25.6	23.4	38.8	34.5	30.6	28.3
	1200	35.0	29.2	23.8	20.7	36.8	30.7	25.7	22.6	43.0	37.2	32.3	30.4	50.5	44.8	37.3	35.7
	1600	44.0	36.7	29.9	25.6	45.8	38.2	31.7	28.1	54.6	47.2	40.4	36.3	62.8	55.8	48.4	45.3
1600	200	16.5	13.5	10.9	10.2	17.1	15.3	13.6	12.3	18.0	16.2	14.2	13.6	19.1	17.1	15.2	14.0
	400	18.2	14.8	12.1	11.2	19.0	17.0	15.0	13.6	20.2	18.0	16.1	14.9	20.9	18.7	16.8	15.4
	800	23.3	19.0	15.7	14.5	24.7	22.1	19.2	17.2	26.2	23.4	20.5	18.9	27.0	24.1	21.7	20.3
	1200	30.6	25.0	20.5	17.9	31.8	28.5	24.6	22.2	33.5	30.0	26.6	24.3	35.4	31.7	28.4	28.9
1200	200	11.3	10.1	9.0	8.3	12.2	10.9	10.0	9.3	12.6	11.3	10.5	9.9	12.9	11.5	10.9	10.4
	400	12.5	11.3	10.4	9.5	14.2	12.6	11.4	10.6	14.4	12.9	12.0	11.2	15.2	13.6	12.3	11.8
	800	16.5	14.8	13.0	12.0	18.5	16.5	15.1	13.7	19.3	17.3	15.8	14.7	20.9	17.9	16.8	15.5

TABLE 6. Wall Temperatures

Gas Temp. F	Stock Temp. F	s = 1 ft Ratio *R* 0.1	0.3	0.5	0.7	s = 2 ft 0.1	0.3	0.5	0.7	s = 3 ft 0.1	0.3	0.5	0.7	s = 4 ft 0.1	0.3	0.5	0.7
2800	200	2140	1935	1735	1580	2280	2055	1865	1725	2410	2180	1965	1830	2465	2205	2000	1905
	400	2170	1945	1740	1600	2300	2070	1875	1740	2425	2190	1980	1850	2480	2225	2025	1925
	800	2190	1960	1765	1650	2325	2090	1890	1760	2440	2200	2000	1885	2510	2250	2055	1950
	1200	2230	2010	1855	1745	2375	2140	1960	1850	2475	2230	2055	1940	2530	2275	2100	2000
	1600	2350	2125	2000	1920	2450	2230	2090	2000	2520	2300	2165	2080	2560	2340	2200	2120
	2000	2450	2295	2210	2170	2525	2365	2270	2215	2550	2420	2325	2260	2600	2450	2350	2295
	2400	2595	2525	2485	2470	2620	2560	2510	2490	2640	2580	2540	2505	2670	2600	2550	2520
2400	200	1820	1715	1595	1440	1960	1835	1680	1570	2100	1930	1755	1645	2135	1995	1820	1680
	400	1830	1740	1615	1460	1980	1860	1700	1600	2115	1940	1775	1670	2155	2005	1830	1700
	800	1850	1770	1640	1500	2000	1890	1730	1630	2125	1950	1800	1715	2170	2015	1845	1730
	1200	1940	1840	1740	1620	2120	1930	1790	1705	2180	2000	1865	1760	2210	2025	1900	1815
	1600	2030	1860	1870	1820	2195	2030	1940	1880	2235	2075	1985	1920	2265	2100	2015	1950
	2000	2200	2140	2105	2085	2265	2190	2140	2115	2285	2210	2160	2135	2310	2225	2180	2150
2000	200	1615	1495	1335	1260	1680	1575	1415	1325	1790	1650	1520	1420	1845	1715	1605	1450
	400	1630	1525	1385	1290	1690	1590	1445	1350	1800	1670	1545	1450	1850	1730	1625	1485
	800	1650	1560	1430	1350	1705	1610	1485	1390	1818	1690	1575	1490	1865	1750	1650	1570
	1200	1735	1640	1550	1510	1780	1675	1580	1525	1860	1730	1650	1590	1900	1780	1705	1650
	1600	1820	1785	1740	1710	1850	1800	1760	1730	1920	1840	1790	1760	1970	1870	1810	1785
1600	200	1320	1250	1180	1000	1410	1310	1200	1120	1470	1350	1250	1170	1495	1380	1275	1215
	400	1330	1275	1200	1075	1435	1330	1240	1170	1490	1370	1275	1200	1505	1400	1300	1240
	800	1365	1310	1230	1170	1455	1350	1290	1230	1505	1390	1320	1260	1525	1420	1340	1285
	1200	1470	1420	1370	1335	1525	1450	1405	1375	1600	1470	1420	1390	1630	1480	1440	1360
1200	200	1050	950	870	840	1105	1015	940	900	1130	1030	955	890	1175	1045	990	945
	400	1100	1000	920	880	1140	1040	970	925	1155	1060	1000	950	1205	1070	1020	975
	800	1160	1070	1010	980	1210	1090	1040	1020	1220	1100	1060	1030	1250	1110	1070	1050

to be opaque, it radiates like a solid body, and the wall assumes the temperature of the flame.

It is again emphasized that Tables 5 and 6 do not take care of three additional variables, namely emissivity of flames, emissivity of heating stock, and variation in coefficient of heat transfer by convection. Figure 47 illustrates the effect of emissivity of both stock and flame on the coefficient of heat transfer for one single set of conditions; namely, where temperature of the gases is 2375 F and of the stock 1500 F, where R is 0.3 and s is 4 ft. It will be seen that a variation in either of these values from the emissivity values of 0.9 for stock and 0.21 for gases used in the preparation of the tables has a pronounced effect on the overall coefficient. If these additional variables were to be embodied in the tables, these tables would become a book. It must also be noted that, in practice, furnaces have not the ideal shape upon which Tables 5 and 6 were based. As a matter of fact, shapes of furnaces have so great an influence that the values of Tables 5 and 6 may be misleading. These facts are discussed in the following chapters.

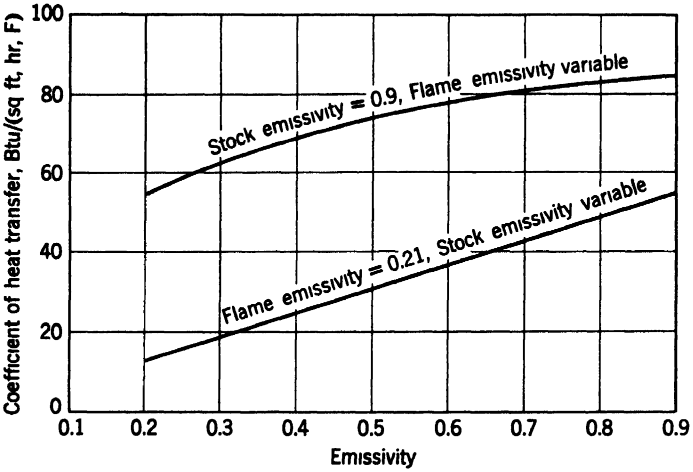

Fig. 47. Effect of emissivity of stock and of flame on coefficient of heat transfer, when $T_g = 2375$ F, $T_s = 1500$ F, $R = 0.3$, and $s = 4$ ft (see Table 5).

chapter 3

The Heating Capacity of Batch-Type Furnaces

1. DEFINITION OF HEATING CAPACITY

The heating capacity of a furnace is usually expressed by the weight of metal that can be heated in unit time to a given temperature at the coldest spot of the charge and without over-heating the rest of the charge. Since the cost of a furnace is approximately proportional to its size, heating capacity per unit volume is important. Specific heating capacity is expressed either by

Weight heated/(hour × furnace volume), or by
Weight heated/(hour × hearth area).

Neither ratio is a perfect measure of heating capacity, which fact is established by the following two examples: In the annealing of huge tanks, the volume of the furnace must be large enough to house the tank and to leave room for circulation of products of combustion. In consequence, the heating capacity per unit of volume is small. If a long gun tube is suspended in a cylindrical annealing furnace, the annealing capacity per unit of hearth area is very great. In the latter case, engineers help themselves by calling the area (furnace diameter × length) the hearth. The criterion: Hourly weight per square foot of hearth is more frequently used than that of hourly weight per cubic foot of volume.

The heating capacity depends on several factors, such as rate of heat release, rate of heat transfer to the charge, and rate of heat conduction to the coldest point in the charge.

2. EFFECT OF RATE OF HEAT LIBERATION

This subject was not discussed in Chapter 2 because the rate of heat release varies with the type of heat-generating equipment.

In electric heating, either by direct resistance or by induction, heat is generated in the material of the heating stock. Induction heating is schematically represented by Fig. 24. Neither of these heating processes requires a furnace, for which reason they do not properly belong in the present volume. They are discussed in Volume II.

The majority of electrically heated furnaces are equipped with heating elements that are fastened to the walls and the roof. The rate at which heat can be released per square foot of (element-covered) wall area depends upon a number of factors, the most important of which is economic length of life of the elements. That length of life is a function of material of the elements, design and spacing of the elements, furnace temperature, and furnace atmosphere. These details are discussed in Volume II. For that reason, the problem of heat release in unit time is limited to fuel-fired furnaces in this chapter.

In such furnaces, heat release is usually expressed in heat units released in unit volume in unit time. In the practice of steam generators, the unit commonly used is: Btu per (cu ft, hr). With this unit, the figures are uncomfortably large. For that reason, Btu per (cu ft, sec) was adopted for this volume. Closely related to rate of heat release is the term "flame length." For open-hearth furnaces and glass tanks, flame length means the distance between the burner and the point where the flame is "extinguished" (becomes non-luminous). For industrial furnaces, flame length means the distance between the burner and the point at which combustion is about 95 per cent complete. Under average furnace conditions, combustion does not go to completion unless excess air is supplied, and metal heating furnaces are seldom supplied with appreciable amounts of excess air, except at low temperatures, or when heating high-carbon or stainless steels. The norm of about 95 per cent completion of combustion also applies to combustion volume. Depending upon type of fuel and design of combustion device, flame lengths range between 1 inch and 40 feet.

Combustion volume and flame length are illustrated by Figs. 48 and 49. The illustrations show geometrically similar gas burners. The large burner (Fig. 48) produces a long flame. If a single large burner is located in the center of a furnace wall, the space surrounding the "flame" is wasted, so far as combustion space is concerned. Contrariwise, many small burners (Fig. 49) utilize the whole wall area, and work with short flames. In Figs. 48 and 49, air is shown as mixing with the fuel at the mouth of the burner. If fuel and air are mixed ahead of the burner (premixed), combustion occurs more rapidly and less combustion volume is needed than with furnace mixing. And if, finally, the combustion air is preheated, combustion proceeds even more rapidly.

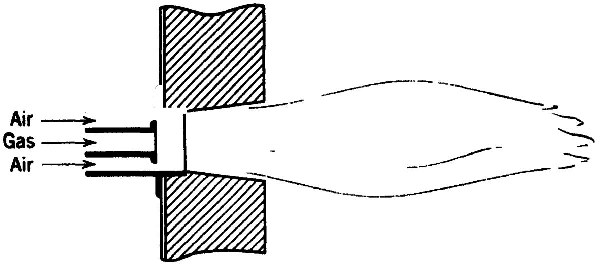

Fig. 48. Poor utilization of combustion space by one large gas burner.

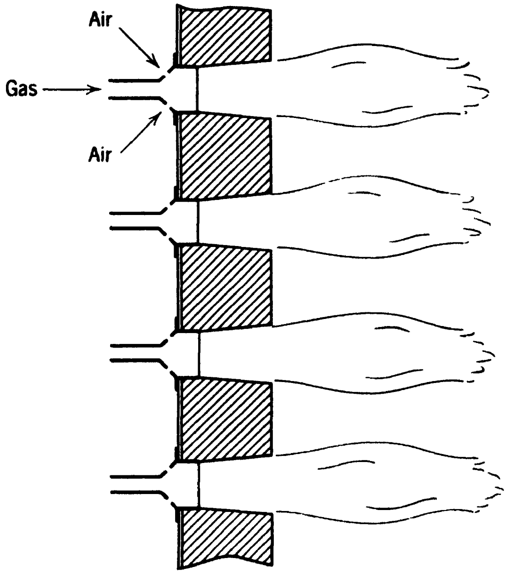

Fig. 49. Good utilization of combustion space by four small burners, same capacity as Fig. 48.

In view of the great number of variables that affect the speed of combustion, it is difficult to predict the volume that is needed to

(95 per cent) complete combustion. For a rough estimate, the information in Table 7 is helpful.

The term "combustion condition" represents combinations of different elements and requires some judgment for its proper use. The following explanation may serve as a guide:

Combustion Condition 1. Very poor mixing of fuel and air, coarse fuel, cold air, inclusion of space in which no combustion takes place under the name "combustion volume."

TABLE 7

Combustion Condition	1	2	3	4	5	6
Btu per (sec × cu ft of combustion space)	1.5	6	10	18	33	500

Combustion Condition 2. Fair (to poor) mixing of fuel and air, fair utilization of volume of combustion chamber, coarse fuel, cold air. Also, same conditions as in condition 1, except 500 F air.

Combustion Condition 3. Good mixing of fuel and air, good utilization of combustion space, very fine atomization or powdering of fuel, cold air. Also, same conditions as in condition 2, but 500 F air.

Combustion Condition 4. Thorough mixing (or premixing) of fuel and air, perfect utilization of combustion space, very fine atomization or powdering of fuel, 500 F air. Also, same conditions as in condition 3, except 1000 F air.

Combustion Condition 5. Thorough mixing (or premixing) of fuel and air, perfect utilization of combustion space, very fine atomization of fuel, 1000 F air. Also, discharge from many small burners.

Combustion Condition 6. Premixed fuel and air issuing from closely spaced small orifices are projected against refractory surfaces, which act as catalyzers and speed combustion. In the combustion space proper, as many as 1000 Btu per (cu ft, sec) are released. To prevent local overheating of the charge, distance must be provided between burners and charge. In consequence, the heat release per unit of vacant (not occupied by charge) volume is a small fraction of the given rate.

For coal burned on the grate, the combustion space is the sum of the volume of the fuel bed and of that volume above the fuel bed in which the combustible gases arising from the fuel bed burn to 95 per cent of completion of the combustion reaction. That volume varies greatly with the amount of volatile matter in the coal, the arrangement of the grate or stoker, and the rapidity with which the secondary air and the combustible gases are mixed above the grate. In industrial furnaces the high rate of combustion obtainable in boiler furnaces having cooler surfaces above and around the fuel bed cannot be utilized, because the roof over the grate or stoker would quickly burn out. Instead, the endeavor is made to effect "semi-gas firing," making the fuel bed act to some extent as a gas producer, and burning the resulting gases over the hearth. In any event, the term "heat developed in unit time and in unit combustion volume," is a misnomer, if coal is burned on a grate or stoker, because only carbon monoxide, hydrogen, and hydrocarbons are burned in the so-called combustion space, whereas the solid carbon is partly gasified in the fuel bed.

The significance of the values in Table 7 is best illustrated by an example: In heating steel to 2200 F on the hearth, a rate of 80 lb per (sq ft × hr) is high. By the use of Fig. 27 it is found that $\frac{80 \text{ lb} \times 340}{3600}$ = about 8 Btu per sec are needed for each square foot of hearth. Depending upon the design of the furnace, somewhere between twice and thrice that quantity of heat must be released. This means that between 15 and 20 Btu must be released in each second per square foot of hearth. With combustion condition 5, about ½ cu ft would be needed per square foot of hearth, and with combustion condition 6, less than 1/25 cu ft would be required per square foot of hearth. The conclusion is that, in fuelfired furnaces of the batch type with proper burners, the rate of heat release is not one of the factors that limit the heating capacity of the furnaces.

In addition to the quantity of heat that is released in unit volume and in unit time, the temperature at which it is released affects furnace capacity. Whereas it is true that each molecule burns at the ideal (adiabatic) flame temperature, the reaction heat is transferred to surrounding objects (gaseous, liquid, and solid), while combustion proceeds. Only by infinitely rapid combustion, or by combustion in a heat-tight chamber, can the adiabatic flame temperature be reached. And even that temperature may not be high enough with very lean fuels. Since heat transfer requires a temperature difference, the furnace would have to be infinitely large if the adiabatic flame temperature equaled the temperature to which the stock is to be heated. This observation is of importance if lean fuels, such as blast furnace gas and clean producer gas, are to be burned. Values for adiabatic temperatures are given in Chapter 5. With lean fuels, high temperatures can be obtained only by preheating the air or the fuel, or both. Oxygen-enriched air likewise increases flame temperature.

3. EFFECT OF RATE OF ABSORPTION OF HEAT BY THE CHARGE

Since ample heat can be released at sufficiently high temperatures in industrial furnaces, the problem of calculating furnace capacity can be studied on the basis of heat transfer to the charge and temperature equalization within the charge.

With adequate heat release at sufficiently high temperature assured, the following variables affect furnace capacity:

1. The temperature of the furnace walls when cold stock is charged.
2. Temperature to which stock is to be heated.
3. Temperature of products of combustion.
4. Emissivity of products of combustion.
5. Emissivity of walls.
6. Emissivity of heating stock.
7. Ratio of wall surface to surface of stock.
8. Thickness of layer of products of combustion.
9. Thermal conductivity of stock (including voids).
10. Desired uniformity of temperature in product.
11. Thickness of heating stock.
12. Size of furnace (hearth area and volume).
13. Velocity of gases in furnace.

There are additional, but minor, items which affect the heating capacity.

It is hopeless to compress these variables into a single equation for heating. Engineers have computed tables and have drawn charts for combinations (of the variables) that fit the types of furnaces and materials that frequently occur in their practice. Tables 5 and 6 are an example of such calculations for one specific configuration of furnace and charge, of one emissivity of refractories, one emissivity of stock, and one composition of products of combustion. It is not the purpose of this book to furnish charts for every conceivable combination. Instead, there will be developed a reasonably simple method that suffices for practical purposes.

Generally speaking, the transfer of heat to the charge determines the heating capacity for thin heating stock, whereas temperature equalization within the stock determines heating capacity for thick stock having low thermal conductivity.

For thin stock having high conductivity, the differential equation of heat transfer is at any time

$$\overset{\substack{\text{Trans-}\\\text{mitted}\\\text{heat,}\\\text{Btu}}}{dQ} = \overset{\substack{\text{Heat}\\\text{trans-}\\\text{mitted/}\\\text{(sq ft,}\\\text{hr, F)}}}{k} \times \overset{\substack{\text{Exposed}\\\text{area of}\\\text{stock,}\\\text{sq ft}}}{A} \times (\overset{\substack{\text{Temp.}\\\text{of}\\\text{furnace}\\\text{gases,}\\\text{F}}}{T_g} - \overset{\substack{\text{Temp.}\\\text{of}\\\text{stock}}}{T}) \times \overset{\substack{\text{Time,}\\\text{hr}}}{dt} = \overset{\substack{\text{Weight}\\\text{of}\\\text{stock,}\\\text{lb}}}{W} \times \overset{\substack{\text{Spec.}\\\text{heat}\\\text{of}\\\text{stock,}\\\text{Btu/}\\\text{lb F}}}{C_p} \times \overset{\substack{\text{Temp.}\\\text{rise}\\\text{of}\\\text{stock}}}{dT} \qquad (8)$$

If k and T_g remain constant during the heating process, equation 8 can be solved by integration, which furnishes the heating time t:

$$t = \frac{WC_p}{kA} \ln \frac{T_g - T_i}{T_g - T_f} \qquad (9)^*$$

where T_i and T_f are the initial and final temperatures of the surface of the stock.

If the temperature of the stock at the end of time t is wanted, the equation is transformed to

$$T_f = T_g - (T_g - T_i)e^{-\frac{kAt}{WC_p}} \qquad (10)$$

It is difficult accurately to measure the temperature T_g of the hot gas. It is easier to measure the temperature T_w of the refractories. This temperature is often (although incorrectly) called furnace temperature. If T_w is used in equation 9, instead of T_g, the value of k changes.

Example for the Use of Equation 9. The largest steel plate that can be accommodated in a furnace having 8 ft by 9 ft hearth area is 6 ft by 8 ft. This plate, which is 2 in. thick, is to be heated from 100 F to 2200 F by products of combustion that have a temperature of 2400 F. How long does it take to heat the plate, and what is the furnace capacity in pounds per square foot, hour?

From information that is given later in this chapter, the average coefficient of heat transfer is found to be 30 Btu per (sq ft, hr, F). Specific heat of steel in the desired temperature range is 0.16.

$$\frac{T_g - T_i}{T_g - T_f} = \frac{2300}{200} = 11.5$$

The log to the base e of this ratio is 2.44. With $\frac{WC_p}{kA} = 0.43$ hour, the heating time is $2.44 \times 0.43 = 1.05$ hr.

Since 1 sq ft of 2-in. plate weighs approximately 80 lb, the heating capacity per square foot of plate is $\frac{80}{1.05}$ = practically 80 lb per (sq ft, hr). The heating rate per square foot of hearth area is $\frac{2}{3} \times \frac{80}{1.05} = 51$ lb per (sq ft, hr).

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* ln signifies "natural logarithm." The symbol formerly used was $\log_e$. "e" equals 2.7183. Either symbol means logarithm to the base "e." The natural logarithm equals 2.3 times the common logarithm.

In the example, a high temperature of the products of combustion is permissible because the furnace charge consists of a single thin piece only, which can be removed when heated to the desired temperature. If the batch consists of several pieces, high excess temperatures are not permissible, as explained below. The sole purpose of this example is to illustrate the use of the logarithmic equation.

Equation 9 may be used to follow the rate of increase of the temperature of the stock during the heating process, even if k varies with temperature. This can be most accurately accomplished if the total temperature rise is divided into several steps. The average coefficient for each step is read from Table 5, Chapter 2,* the individual heating times are computed from equation 9, and are added. Subdividing the temperature rise into steps promotes accuracy, because the variation of k within each step is small.

The curves of Fig. 50 are the result of such calculations for the same condition as applied in the preceding example, and also for a flame temperature of 2800 F. It can be quickly recognized that the higher temperature of the products of combustion results in a shorter heating time. In the illustration, curve *B* refers to 2800 F. Figure 50 brings out another fact: Whereas high temperatures of gases and walls reduce heating time, these high temperatures call for prompt removal of the stock, to prevent overheating.

Heating time and heating capacities obtained from the use of equation 9 extending over the whole temperature range are correct if the coefficient of heat transfer remains constant while the charge is being heated. This requirement is approached, but not entirely met, in ovens and in low temperature furnaces, that is to say within the range from 800 F to 1200 F, because heat transfer by convection is an important factor at these low temperatures, which means that the coefficient k does not vary as much during the heating process as it varies in high-temperature furnaces. In consequence, an equivalent constant coefficient k can be used in equation 9. The introduction of an average coefficient carries with it an (although small) uncertainty and approximation. For that reason and also because engineers dislike the logarithmic equation, a simpler and equally approximate method may well be adopted, which is sufficiently accurate for all practical purposes.

Heat transfer is proportional to the product: difference (furnace temperature minus average temperature of stock), times average coefficient of heat transfer. The determination of both averages is discussed in the following paragraphs.

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* Table 5 is correct if the idealized conditions prevail for which it was calculated. In most industrial furnaces, the coefficients are about 70 per cent of those given in Table 5. They are even lower if stock with low emissivity is heated (see Fig. 52).

In a furnace of the batch-type, temperatures of products of combustion and of refractories must be so selected that the charge is not injured by overheating, if it stays in the furnace during an unusually long time. This requirement fixes the temperature of the products of combustion to be about 5 per cent higher (from zero F, not absolute) than the prescribed final surface temperature of the charge. The excess temperature may be higher, if occasional overheating causes no serious damage to the heating stock. In that case, the before-mentioned 5 per cent are sometimes increased to 8 per cent.

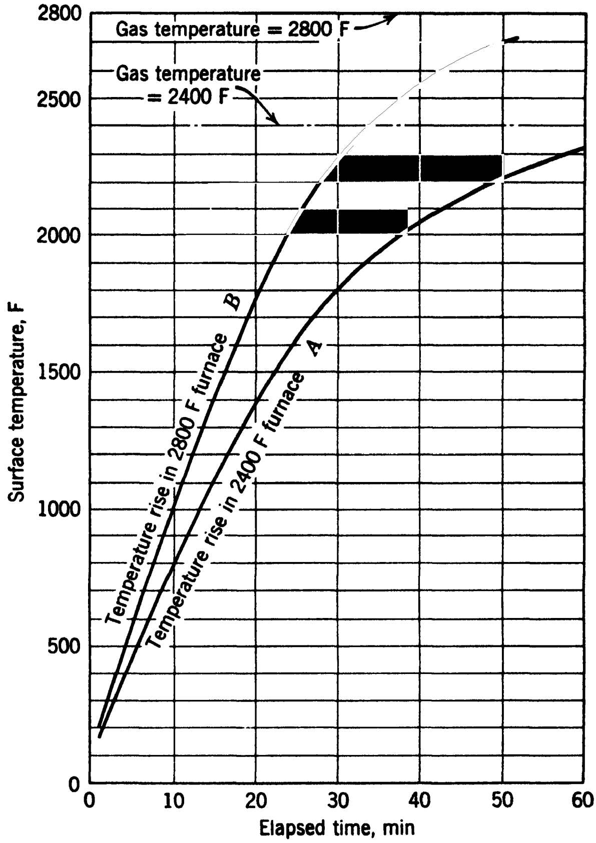

Fig. 50. Heating curves for steel plate 2 in. thick, heated from top only. Ratio $R = 0.5$; gas blanket thickness $s = 2$ ft.

Depending upon the selected excess temperature of products of combustion, the time required for heating is based on 92 to 98 per cent of the available temperature difference. In all cases, the average temperature of the surface of the stock, with regard to time, is 71 per cent of the temperature rise if the heat is transferred by convection only. It is 65 per cent, if the heat is transferred solely by radiation. A value of two-thirds of the temperature rise is suitable for quick calculations.

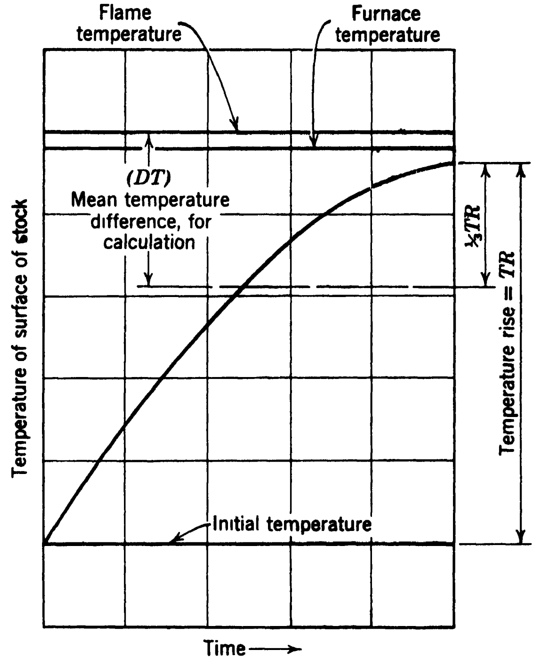

Fig. 51. Diagram of time-temperature relations for calculation purposes.
DT = mean temperature difference
TR = temperature rise

The time-temperature relation is illustrated by Fig. 51. The shape of the curve is not invariably fixed. It varies with the ratio of stock surface to surface of refractories, because furnace temperature drops when a cold charge is deposited in the furnace. The time-temperature curve varies with the thickness and conductivity of the stock, and, as already mentioned, with the ratio of heat transferred by radiation and convection. In a charge of several pieces, the first piece is sometimes removed before it has reached the desired temperature, whereas the last piece has reached furnace temperature at the time of removal. In spite of these variations, the value of $\frac{2}{3}TR$ is safe for design purposes.

By the use of this average the example may be calculated without the use of the logarithmic equation 9. In that example of heating a 2-in. plate to 2200 F, radiation predominates. Hence, the active temperature difference equals $\frac{1}{3}(2200 - 100) + 200 = 900$ F. Then the heating time equals

$$\frac{80 \times 0.16 \times (2200 - 100)}{30 \times 900} = 1.0 \text{ hr}$$

It should again be noted that the high excess temperature (exceeding 4 per cent of the available temperature difference) is permissible in the case of the example, because only one piece is being heated, and it can be removed when the desired temperature has been reached. In the majority of batch-type furnaces that condition does not exist.

The average coefficient of heat transfer (average during the heating process) is a function of many variables, the most important of which is the temperature to which the stock is to be heated. For that reason, it is appropriate to plot the coefficient of heat transfer as a function of the final temperature of the charge or stock. Among the other variables, the ratio *R* of exposed surface area of stock to interior area of refractories is important. Whenever a cold charge is put into a hot furnace, the temperature drops. If *R* is small, the temperature drop is small. If *R* is small, the heating time is reduced. Another important variable is the emissivity of the stock. All three variables and their influence on heating time were combined in Fig. 52.

This figure merits discussion. The values were, in part, derived from Table 5. A comparison of Fig. 52 with Table 5 shows that the coefficients of the figure (which agree with furnace practice) are lower than those in the table. It is interesting to investigate the reasons for the difference. Table 5 was calculated for emissivities of refractories and of stock equal to 0.9. Tests of emissivities of refractories at high temperatures are scarce. The few available tests indicate that the emissivities at high temperatures are generally lower than 0.9. When cold stock is put into the furnace, the temperature temporarily drops by heat imparted to the stock and by radiation through open doors. Some parts of the refractories have lower temperatures than is indicated by the temperature control equipment.

Evaluation of these effects and comparison with existing furnaces resulted in the curves of Fig. 52, which are approximately 30 per cent below the values of Table 5.

From Fig. 52, it follows that heating time is lengthened if *R* is large. In most industrial furnaces, *R* is about 0.4. In the use of higher values of *R*, care must be exercised. If, for instance, square bars or

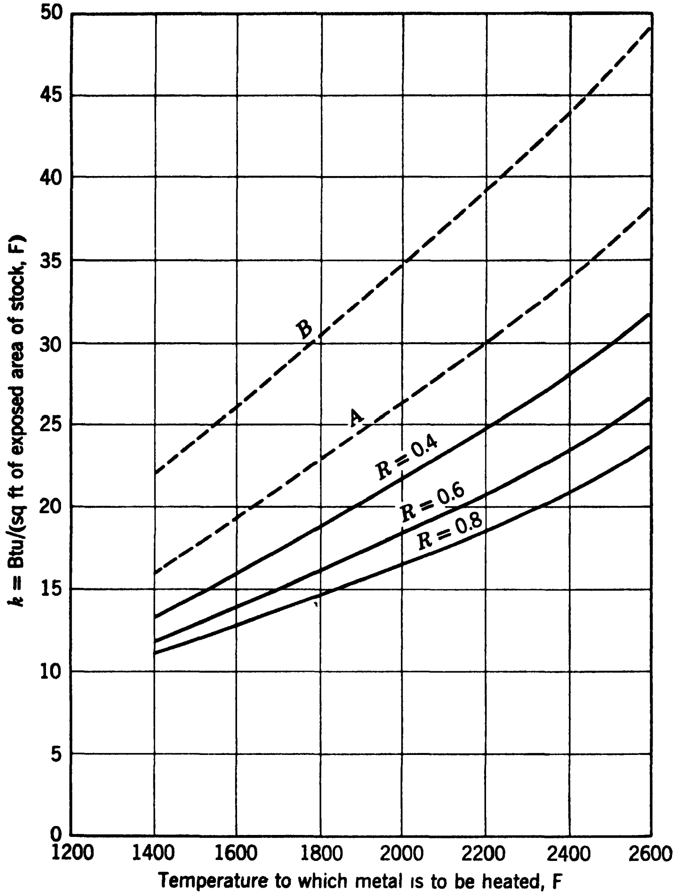

Fig. 52. Overall coefficient of heat transfer to surface of stock.

R is ratio of stock area to area of refractories in furnace (see text).

The curves were derived for emissivity of refractories of 0.9 and stock emissivity of 0.9; multipliers for other stock emissivities are:

For stock emissivity of 0.4, multiply values by	0.43
0.6	0.65
0.8	0.86

For luminous flames, values of the coefficient are approximately 10 per cent higher than shown.

When applying coefficients of this chart, use mean temperature difference indicated in Fig. 51. Values do not apply when the furnace and charge are heated up together. Read p. 60 before using.

For use of curves *A* and *B*, see Chapter 4. Curve *A* applies to continuous heating furnaces where the flame temperature exceeds the final stock temperature by approximately 15 per cent. Curve *B* applies to axial heating.

blocks are set close together, then it is incorrect to base *R* on the total of the top and two sides of the blocks.

Values of *R* that are lower than 0.4 slightly reduce heating time, but do not utilize the full heating capacity of the furnace, because that capacity is related to the product $k \times R$.

If the flame is luminous throughout the length of its travel, *k* is approximately 10 per cent higher than indicated by Fig. 52. And if thick stock is to be heated rapidly to a temperature in excess of the desired final temperature with subsequent drop to soaking temperature, the coefficient *k* may be selected for a temperature between highest temperature and final temperature. Such departures from the values of Fig. 52 require judgment.

4. EFFECT OF ARRANGEMENT OF STOCK

In furnaces of the batch-type, individual pieces (or a group of individual pieces) are heated. The problem arises: What is the effect of arrangement and thickness of the pieces on furnace capacity?

Obviously, space must be provided between the pieces for the manipulating tongs. Unless the spaces between the pieces are inordinately large, the heating capacity is not noticeably affected, because the bare spots of the hearth receive radiation from the gases as well as from the roof and the side walls. The heat received by the hearth is radiated to the work and assists in heating it. If, for instance, the 3-in. billets illustrated in Fig. 53 are laid on the hearth in such a manner that the width of each empty space equals the width of each billet, then the weight per square foot of hearth is the same as if the hearth were covered by a plate or slab 1½ in. thick. The heating surface of the billets is 50 per cent larger than the heating surface of the slab. However, the vertical heating surfaces are not so effective as the horizontal heating surfaces. The subtended angle of radiation is only 63½ degrees, or 127 degrees for the two vertical surfaces. Radiation from the hearth (which is not as hot as the roof) increases the transfer

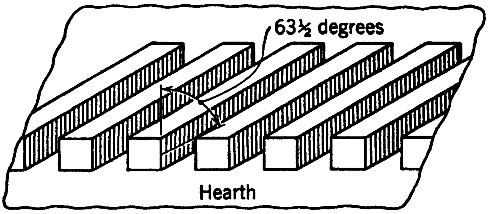

Fig. 53. Billets spaced apart on furnace hearth.

of heat to the vertical surfaces. The result is that the weight of billets heated in unit time equals or even slightly exceeds the rate at which the 1½-in. plate would be heated.

This example proves that the factor R must be used with caution. If R is 0.4 for the plate, then it would be 0.6 for the billets. Inconsiderate use of Fig. 52 would result in a lower value of k for the billets than for the plate, whereas, actually, they are equal.

A similar non-mathematical reasoning may be employed in other cases of seemingly abnormal arrangements of the charge. An example is furnished by the arrangement that is illustrated in Fig. 54. Space for tongs is provided between adjacent blocks. In consequence, the pieces (1) receive heat from three sides in addition to top and bottom. Row (1) heats somewhat more slowly than it would if the blocks had been laid on a hot hearth. After the blocks (1) have been removed, one by one, the blocks in row (2) heat more rapidly than they would if lying on the hearth, because they received some heat while row (1) was being heated. Evidently, a larger stock-covered hearth area would be required, if the blocks were laid flat on the hearth. However, the charging arrangement of Fig. 54 requires a wide strip of uncovered hearth in front of row (1), for the purpose of providing a thick layer of radiating gases.

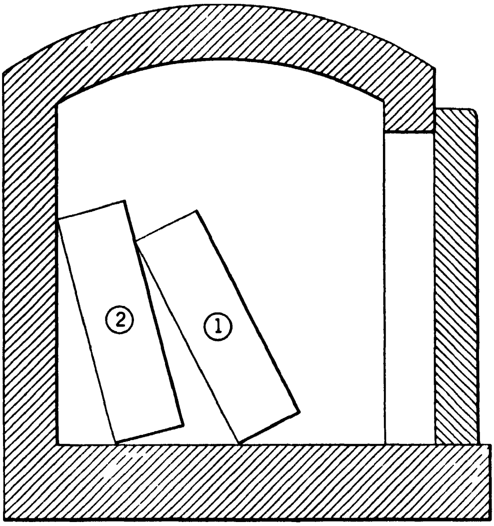

Fig. 54. Furnace with two rows of billets.

The charging arrangement shown in Fig. 54 is common in furnaces that serve forging presses. For continuity of operation, each press is surrounded by several furnaces, usually 3 but sometimes 4. One furnace is being charged, another furnace is heating up, the third furnace serves for soaking, and the fourth furnace is being worked out.

Another abnormal arrangement of the charge is illustrated by Fig. 55. The belief that the three-high packing increases the heating capacity in pounds per (sq ft, hr) is wrong. It takes more than three times as long to heat the stack than is required to heat a single layer.

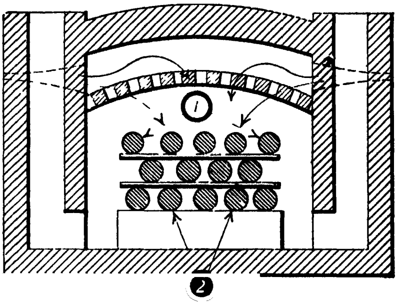

Fig. 55. Effect of arrangement of material on heat transmission.

The pieces marked (2) are heated indirectly by radiation from the pieces in the top row (1) and by almost stagnant products of combustion. The method of stacking the charge is acceptable, if the bars in the upper row are withdrawn, one by one, followed by the bars in the middle row and finally by those in the bottom row. The heating capacity is not improved over that with but a single layer. The advantage consists in carrying on the hot-forming process a longer time without recharging the furnace. Stacking is permissible and even advantageous in heating to low temperatures by forced circulation. See below.

5. EFFECT OF THICKNESS OF STOCK

The stacked charge leads to a discussion of the effect which thickness of charge or stack has upon furnace capacity. The heating of metals being an old art, rules have been formulated on the relation between thickness of stock and time required for heating to a reasonably uniform temperature. For steel, the following rules are in use. Heat travels ⅛ in. in 5 min. Another rule: Cylinders or square bars must be heated at the rate of 20 min per inch of diameter, if of mild steel; 40 min per inch of diameter are required for alloy steel. For annealing of steel 60 minutes are needed per inch of diameter or thickness, if the stock receives heat all around, and if the stock is relatively small with respect to the size of the furnace. These rules were derived from the practice of forging, rolling, and annealing steel. The first rule means that a plate which is 3/8 in. thick requires 3 × 5 = 15 min if heated from one side. The thumb rules are based upon an acceptable temperature difference between surface and core (coldest spot) of the stock. The relation between diameter of stock and temperature difference is illustrated by Fig. 56. For equal heat flow into unit surface, the temperature gradients at the surface are equal regardless of diameter of stock. The tangent to the temperature curves is horizontal at the center. The temperature curves are geometrically similar; the temperature difference is proportional to the diameter of the stock. If the temperature difference at the end of the heating period is to be the same regardless of diameter or thickness, then the heating time per inch of diameter or thickness must grow with the diameter, approximately as the 1.2 power of the diameter ratio.

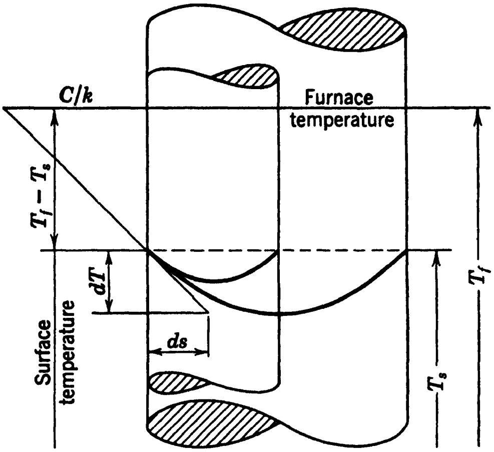

Fig. 56. Temperature relations in geometrically similar bodies.

In the heating of ingots and large forging billets, as well as in annealing, temperature uniformity is reached only after a long time. The time during which the temperatures are equalized is known as the soaking period. If the furnace temperature at no time exceeds the desired final temperature of the charge by more than 2 to 4 per cent of the desired temperature rise, no well defined soaking zone exists.

Even if furnace temperature exceeds final temperature of stock but little, the "core" of the stock is, in large pieces, far from having reached the final temperature. No quick method exists for calculating the soaking period. An equation for computing the soaking time of a cylinder, the heat supply to which is suddenly discontinued, was given by A. Newman and A. Church in the *Journal of Applied Mechanics,* 1935. Use of their equation requires familiarity with

Bessel's functions. Either the Schmidt graphical method or numerical calculations (see Appendix) may be resorted to, but either method is slow. An easily applied method exists for computing the soaking time if the surface temperature of a solid is kept constant during the soaking period. From the curves of Fig. 57* the following ratio can be found.

$$\frac{\text{Temperature difference between surface and center, after } t \text{ hr}}{\text{Temperature difference at beginning of soaking period.}}$$

In order to bring out the effect of size of stock on the soaking period, Fig. 57 was applied to three different long cylinders with the same rate of heat input in Btu/(sq ft, hr). The rate is to be such that, at 2200 F surface temperature, the core temperature of the 3-in. cylinder is 2200 — 40 = 2160 F; that of the 6-in. cylinder is 2200 — 80 = 2120 F; and that of the 12-in. cylinder is 2200 — 160 = 2040 F. If the supply of heat is reduced to such an extent that the surface temperature remains constant at 2200 F, how long will it take to raise the core temperature to 2180 F? In the tabular calculation, j = diffusivity = 0.21 sq ft /hr for mild steel at 2200 F.

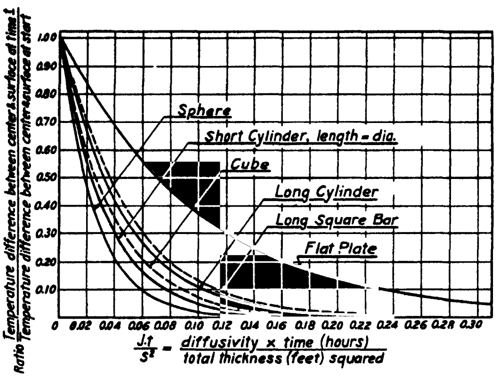

Fig. 57. Temperature-time-space relations during "soaking," with temperature of surface of body held constant.

Diameter of cylinder, ft	¼	½	1
Initial temp. difference DT_i, F	40	80	160
Final temp. difference DT_f, F	20	20	20
DT_f/DT_i	0.5	0.25	0.125
jt/s^2 from curve	0.032	0.064	0.10
Soaking time, t, minutes	0.59	4.6	28.5

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* The curves of Fig. 57 for the plate, long cylinder, and sphere were computed from data similar to those given on pages 439–443 of the Appendix, whereas those for the square bar, cube, and short cylinder were taken from the work of Williamson and Adams, as reported in *Physical Review*, Volume 14, Series 2, page 99.

Obviously, there is no soaking period in the heating of round bars 3 in. in diameter. The soaking time grows almost as much as the cube of the diameters. For an ingot 2 ft in diameter, it runs into hours, if the same small temperature difference is demanded. From Fig. 57, it follows that the soaking time for slabs is longer than for cylinders of equal thickness.

If the volume of a body is small compared to its heat-receiving surface, the body can be heated quickly with but a small difference of temperature between surface and core. For comparison of the effect of different shapes, the plate of large extent, the long cylinder, and the sphere are good examples. If r is the radius of the sphere, also of the cylinder, and also one-half the thickness of the plate, then the ratio volume/surface-area equals r for the plate, $\frac{1}{2}r$ for the cylinder, and $\frac{1}{3}r$ for the sphere. From these values, it may be concluded that, approximately, the heating times for a given temperature difference between surface and center are in the ratio of 1 for the sphere, 1½ for the cylinder, and 3 for the plate. It is premised that the three shapes receive heat uniformly from all sides. The result is only a rough approximation.

In the heating of steel for rolling or forging, practice pays little attention to the fact that soaking time should increase much more rapidly than diameter or thickness of stock, provided the diameter does not exceed 12 in. and heat is received uniformly from all sides. In the heating of larger sections of mild steel, the supply of heat is high in the early stages and is reduced when the surface of the stock approaches the desired final temperature. This method of heating has been (although erroneously) called accelerated heating. Certain alloy steels have a low thermal conductivity below red heat, and are damaged by accelerated heating, as explained below.

Heat supply from all sides is approached by placing the stock on ledges or stools. In many furnaces, the supports are not tall enough to permit equality of heat flow from all directions.

If individual pieces are laid close together on ledges or other supports, heat supply from the bottom and to the sides is obstructed. The fraction of heat which is imparted to the bottom of the charge must be estimated in each separate case, because a correct theoretical calculation is impracticably long.

In the early years of the present century, the wish to improve uniformity of heating and to thereby increase furnace capacity led to the building of underfired furnaces. Figure 7 is a schematic illustration of an underfired furnace. Such a furnace presents problems of strength, and also of circulation of furnace gases. Both problems are discussed in later chapters. In the present chapter, heat supply from below, through the hearth, is the topic.

Underfired furnaces are limited to a maximum temperature of about 1800 F by the strength and durability of the hearth and supports, and the rate of heating of steel in furnaces of this design averages 30 lb per (sq ft, hr), with a maximum rate of about 40 lb.

The heat transferred from the combustion chambers to the heating chamber is the result of (1) radiation through openings along the sides of the hearth, (2) conduction through the hearth, and (3) heat in the gases entering the heating chamber, which for good uniformity are limited to a temperature about 200 F higher than that of the heating chamber. Of these, the heat transmitted through the hearth is a small percentage of the total.

From Fig. 26, the heat content of one pound of steel between room temperature and 1800 F is 305 Btu. A heating rate of 30 lb per (sq ft of hearth, hr) requires the transmission of $30 \times 305 = 9150$ Btu per hr, neglecting the heat losses from the heating chamber. The thermal conductivity of firebrick at furnace temperature equals 9 Btu × in. per (sq ft, hr, F), and the usual hearth thickness is 4 in., so that the heat transmitted through the hearth with a temperature difference of 200 F is $\frac{9 \times 200}{4} = 450$ Btu per (sq ft, hr), or about 5 per cent of the total.

Uniformity of temperature and heating capacity are increased, if the underfired hearth is perforated as illustrated in Fig. 7. In that case, the charge must be placed on high ledges to prevent spotty heating.

The underfired furnace does not increase heating capacity per square foot of floor space, but it serves other useful purposes:

1. The cold spot at the bottom of the charge is eliminated.
2. The temperature of the furnace gases is reduced, and
3. The periphery of the charge is protected from excessive radiation.

As previously mentioned, big objects of certain steels must be heated slowly. Temperature differences produce strains that are harmless in mild steel, but cause cracks in the tender steels. The cracking is accompanied by a peculiar noise that is called the clink. Evidently, slow and careful heating of large objects reduces the heating capacity of a furnace below the average value. The metallurgist sets the heating rate. His orders must be obeyed by the operator of the furnace.

6. VERTICAL HEATING

If elongated objects are to be heated in a vertical position, the term "weight heated per (sq ft, hr)" is meaningless, because the vertical dimensions encountered in practice range from 4 to 60 ft or more. If a single piece, or one row of pieces is heated, engineers help themselves by calling the product, vertical dimension times horizontal dimension of the furnace, the hearth area, to which the rule of weight heated per (sq ft, hr) is applied.

It does not help to lay the furnace on its side in the cases of ingots in a soaking pit and coil annealing. In ingot-heating furnaces (soaking pits), two or three rows of ingots are needed. Calculation of heating capacity of such furnaces is tedious and almost impossible, not only because of variation in size and composition of ingots, but also because some ingots are charged cold and others are charged hot. A common rule for heating time is 1 hr for each hour of track time from pouring at the melt shop to charging into the pit, but this time may vary from 1 hr to 8 hr for carbon steel. For some alloy steels, the heating time may be necessarily twice as great as that obtained from this rule. For an ingot heating furnace 8 ft wide by 22 ft long, the charged weight varies from about 72 tons for alloy steels to 140 tons for large ingots of carbon steel. For the usual alloy steels, the production will be about 6 tons of ingots heated per hour (hot ingots at usual temperatures), which is equal to 68 lbs/(sq ft bottom area, hr), or 45 lbs per (sq ft side area, hr). For low carbon steel, the production is from 10 to 20 tons per hr, with a usual average of about 14 tons per hr. Additional complications are introduced when it is necessary to provide sufficient charging capacity to handle wide variation of production at the melt shop, resulting in "bunched heats," and by mill capacity when rolling ingots of different sizes and analyses.

Consideration is given later in this chapter to the heating of vertically piled coils of wire and strip.

7. MUFFLE FURNACES

The principal purpose of a muffle is to protect the charge from the effects of the products of combustion. Another object is to smooth out temperature inequalities that exist on the combustion side of the muffle wall. A muffle that serves only to smooth out temperatures is frequently open and is called a semi-muffle. The muffle may surround the charge, as diagrammatically illustrated by Fig. 58. It may at times surround the burning gases, and when it does, the muffle usually is a tube (radiant tube). (See Fig. 16.)

The heating capacity of a muffle that surrounds the charge depends upon the surface area and emissivity of the stock, and upon properties of the muffle wall (temperature, area, emissivity). For a given set of these variables, the heating capacity is slightly less than that of a direct-fired furnace (having the same wall temperature) because of the absence of products of combustion that are hotter than the furnace wall.

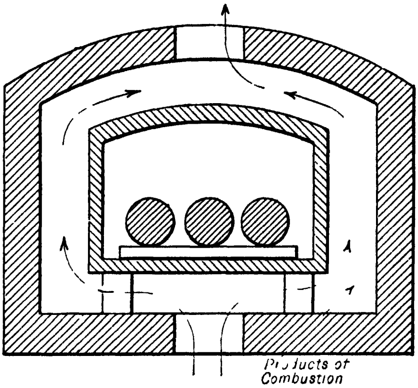

Fig. 58. Muffle furnace.

For muffles that surround the charge, the following facts should be noted. The inside of the muffle wall is colder than the outside, because of thermal resistance of the muffle wall. And the outside of the muffle wall is colder than the refractories that surround the muffle, because a muffle is a radiation shield which reduces heat transfer (due to solid radiation) by one half. Proof is furnished by the following reasoning: Let area A having absolute temperature T_1 radiate to an equal area A having absolute temperature T_2. Then the hourly heat transfer $Q = kA(T_1^4 - T_2^4)$. If an extremely thin wall is interposed between the two areas, that wall quickly assumes an intermediate absolute temperature T_m. The intermediate wall must radiate to the low-temperature area as much heat as it receives from the high-temperature area, in order to be in temperature equilibrium. This fact leads to two intermediate equations, namely

$$Q/(kA) = T_1^4 - T_m^4 \quad \text{and} \quad Q/(kA) = T_m^4 - T_2^4$$

By addition of the two equations, the intermediate temperature cancels out. The resulting equation, $Q = \frac{1}{2}kA(T_1^4 - T_2^4)$, proves that even an infinitely thin muffle cuts heat transfer due to solid radiation in half. However, flame radiation does not follow the fourth-power law; this means that the reduction is slightly less than half.

The capacity of a muffle is also limited by strength and durability of the muffle walls. This fact can be illustrated by an example. Let it be required to heat steel to 1000 F in the furnace which is illustrated by Fig. 58, and let that steel be heated at the rate of 25 lb per (sq ft, hr). From Fig. 26 or 27, the heat to be imparted to each pound of steel equals 250 Btu. The hourly heat input into the steel per square foot equals 25 × 250 = 6250 Btu. For average conditions, the ratio of heat-transmitting muffle surface to hearth area is about three. With this value, each square foot of muffle surface has to transmit 2080 Btu per hour. If the muffle is made of fireclay, which has an average conductivity of 9 Btu in. per (sq ft, hr, F), and if the muffle wall is 1 in. thick, the temperature drop through the muffle wall would, in the case of the example, equal 2080/9 = 232 F. Obviously, if the muffle were 2 in. thick, the temperature drop would be 464 F; and it would be only 58 F, if the muffle were ¼ in. thick. In Chapter 7, proof is offered that cracking of fireclay tiles cannot be prevented if the number of Btu transmitted through the wall per square foot and hour, multiplied by the thickness in inches, exceeds a certain value. That value is about 2000 for tiles made with great care of selected material, and is low as 500 for tiles made with less care from poorer material. Translated into the present example, it means that the best possible muffle wall could be 2000/2083 = 1.0 in. thick, whereas a muffle of poorer quality could be only 500/2083 = 0.24 in. thick if cracking is to be prevented.

The great difference of temperature on each side of a muffle wall that is made of fireclay not only reduces the heating capacity of the muffle but also causes the products of combustion to leave at a very high temperature. For both reasons, muffle walls are made as thin as is practical of material that has both high thermal conductivity and resistance to heat. Alloy steel and silicon carbide are the most suitable materials for muffle walls.

An example of the capacity of muffle-type furnaces is given at the end of this chapter.

Muffles are seldom used for heating metals to more than 1800 F because of cost of maintenance. For lower temperatures, electrically heated furnaces and furnaces with radiant tubes and forced circulation have taken over.

The heating capacity of furnaces that are equipped with flame-in-tube muffles (radiant tubes) is limited by the heat that can be radiated from the tubes in unit time. Since radiant tubes are more frequently used in continuous furnaces, their radiating capacity is discussed in the next chapter.

8. HEATING CAPACITIES IN TEMPERATURES FROM 1400 F TO 1800 F

The metals and the purposes for which they are heated in this temperature range are given in Table 8. The heating capacity of furnaces that operate within that temperature range can be determined in the manner used above for high temperature furnaces. Although less heat needs to be imparted to each pound of the stock, heating time is longer and heating capacity is lower, because heat transfer by radiation is smaller than it is at higher temperatures. By reference to Fig. 37 (Chapter 2) it can be seen that the coefficient of heat transfer from 1600 F to 1200 F is only 40 per cent of the coefficient for the same difference between 2200 F and 1800 F. The effective temperature difference (between furnace and stock) is also lower, but that decrease is counterbalanced by the lower amount of heat required.

In the temperature range examined here (1400 to 1800 F), the heat transfer coefficients of Fig. 52 apply, if attention is paid to the multipliers. If bright metals such as, for instance, stainless steel or titanium are heated, the rate of heat transfer must be adjusted in conformity with equation 7.

Copper and its alloys are heated within the temperature range that is considered in this section. In comparing the heating time of these alloys with the heating time of steel, and basing the comparison on equal temperature uniformity in the stock, the heating time per inch of thickness of the other metal equals the heating time of steel multiplied by

$$\frac{\text{Conductivity of steel} \times \text{Specific heat of other metal}}{\text{Conductivity of other metal} \times \text{Specific heat of steel}}$$

This multiplier is based on the observation that the rate of heating of the interior of the metal, in cubic feet per hour, is proportional to

$$\frac{\text{Conductivity}}{\text{Specific heat} \times \text{Density}}$$

and the weight heated in pounds per hour is, of course, proportional to the latter ratio, multiplied by the density. Values for computing

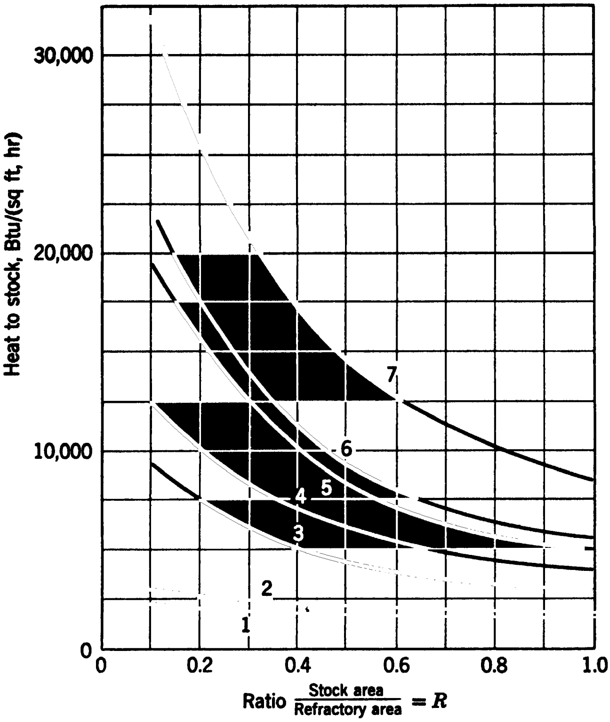

Fig. 59. Overall rate of heat transfer to surface of stock. These average values for the rate are taken from practice, and apply to the following conditions:

Curve 1. Bright aluminum at 900 F—forced convection
2. Oxidized aluminum at 900 F—forced convection
3. Heat treating of steel at 1500 to 1700 F
4. Heat treating of thin steel at 1500 to 1700 F
5. Oxidized copper at 1600 F
6. Forging or rolling of steel at 2100 to 2200 F
7. Forging or rolling of steel at 2300 to 2400 F

When applying coefficients of this chart, use mean temperature difference indicated in Fig. 51. Values do not apply when furnace and charge are heated up together. Read p. 60 before using.

the multiplier are given in Table 3. For copper, the multiplier is about one-ninth, and for brass it is about one-half.

If the above given multiplier were the only criterion for heating capacity, copper could be heated in much smaller furnaces than are needed for heating the same weight of steel. Actually, other factors must be considered, such as emissivity and temperature of refractories (see Fig. 59). The upshot is that at 1600 F about twice as much copper can be heated per square foot of hearth and hour as steel.

9. HEAT TRANSFER AND HEATING CAPACITY AVERAGE VALUES FROM PRACTICE

The danger of overheating the stock limits the capacity of furnaces of the batch-type. When the hearth is properly filled, the ratio *R* of stock surface to refractories surface is not subject to great variations. In consequence, average rates of heat transfer and of heating rates have been established. Figure 59 is an example of such averages. The data, which were collected by the junior author, reveal the relations between temperature to which the stock is to be heated, ratio *R* (stock surface area to area of refractories) and rate of heat transfer. The values of Fig. 59 are averages and are subject to considerable deviations in individual cases. Among the reasons for deviation is the fact that not all of the surface of the stock can see the heated refractories. Part of the stock may see refractories that are below furnace temperature.

TABLE 8

Material and Purpose	Desired Final Temperature	Heating Capacity lb/sq ft, hr (total hearth area)
Steel		
Forming, low carbon	2350	60
Forming, high carbon, alloy	2100	40
Forming, stainless	2300	40
Butt & Lap weld, skelp	2650	90
Hardening	1650	30
Drawing	1000	25
High Anneal	1500	5
Low Anneal or Temper	1250	15
Copper and Brass		
Forming, cake heating	1600	100
Breakdown Anneal, Coils, Sheets, Tubes	1325	70
Finish Anneal, Coils, Sheets, Tubes	1325	35
Aluminum		
Forming, Ingots	1000	65
Forming, Billets	1000	35
Annealing, Coils	900	35

In Table 8, as everywhere else in this book, hearth area refers to the whole hearth. Like Fig. 59, Table 8 offers average values only.

Values of *R* lower than 0.4 help in the quick heating up in a furnace that is too large for the job at hand. Furnaces that are charged to optimum capacity do not exhibit such low values of *R*. In unusual cases, heating capacities of furnaces may deviate as much as 50 per cent from the values of Fig. 59.

Another statistical method consists in expressing furnace capacity by gathering data on pounds of metal heated per (sq ft × hr). Such data, carefully selected by the junior author, are given in Table 8.

10. LOW-TEMPERATURE FURNACES AND OVENS

The study of furnace capacity has, up to this point, been limited to furnaces in which a temperature in excess of 1400 F prevails. In such furnaces radiation is very energetic, so much so that all parts (except where covered by the charge) of a batch-furnace have practically equal temperatures. Matters are different in ovens and furnaces using a low temperature, at which radiation is weak. At 2240 F, unit area radiates 16 times as much heat as it radiates at 890 F. Whereas the heat to be imparted at 890 F is 36 per cent of the heat required by the same weight of charge at 2240 F, the energy of radiation is only 6 per cent. The result is two-fold. If heat were to be transferred by radiation only, the low temperature furnace would, for a given rate of heat transfer, have to be six times as large as the high temperature furnace, and the temperature of the charge would be far from uniform. To make matters worse, the materials which are heated to between 890 and 1200 F are usually bright and, therefore, do not readily absorb radiant heat.

The difficulties are largely overcome by increasing the convection component of heat transfer. From equation 3, in Chapter 2, it is known that the rate of heat transfer by convection grows with greater density and with greater velocity of the flowing medium. At low temperatures, the density is greater than it is at high temperatures. The velocity of the flowing medium is increased, either by the jet action of excess combustion air or, in the majority of cases, by hot fans. The fans may be located in the furnace, where they stir up the atmosphere, but are preferably located in ducts, as shown in Fig. 60. The outlet port or vent is not shown in the illustration. A limit to the velocity is set by the balance between the reduction of furnace size (and cost) and increased temperature uniformity on one side, and cost of power on the other side. The optimum velocity varies with the cost of power and with the condition of the charge, emissivity being one of the factors. The brighter the charge, the higher the optimum velocity. The power delivered to the fan is converted into heat.

Since the heating capacity of any furnace is largely determined by the coefficient of heat transfer, the value of the coefficient must be ascertained. Table 9, which was compiled by the junior author, may be used for this purpose.

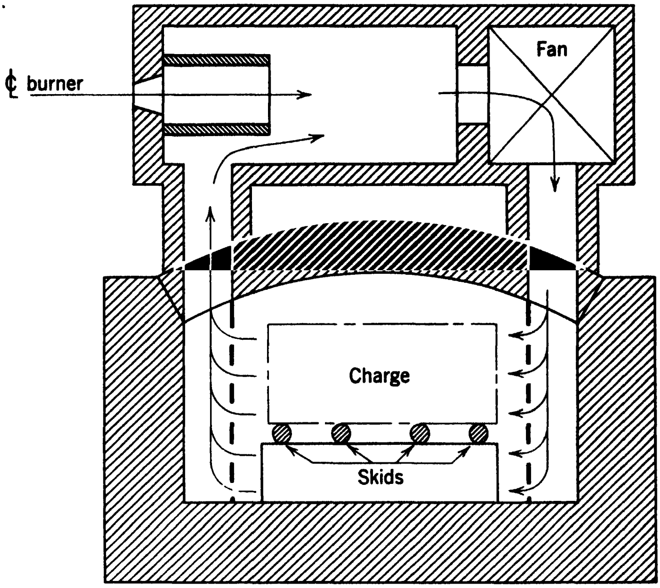

Fig. 60. Forced-convection or recirculating-type furnace, batch type.

Friction losses and power consumption, affecting the selection of the coefficient of heat transfers, are discussed in Chapter 8. After selection of coefficient *k* of heat transfer, the heating time can be calculated at the place where the hot gases first contact the charge. Either the logarithmic equation or the simpler method of figuring with the equivalent temperature difference may be used.

While the furnace gases pass along or through the material that is to be heated, they lose temperature. This drop in temperature raises a question, which may be asked in two ways:

1. When the charge at the place of first contact with furnace gases has reached the desired temperature, what is the temperature of the charge at the point where the gases leave?
2. When the coldest point of the charge has reached the desired temperature, how much is the hot part of the charge overheated?

TABLE 9. Coefficients of Heat Transfer in Low-Temperature Furnaces and Ovens, Btu per (sq ft, F), When Gas Temperature Is 100 F Higher Than the Final Temperature of the Stock

		Radiation				Convection			
		Material Being Heated				Velocity of Gas, ft per sec			
Temperature of Gas, F	Stock area / Refractory area R	Oxidized Steel or Copper	Bright Steel or Copper	Oxidized Aluminum	Bright Aluminum	20	40	60	80
1400	0.4	16.2	8.1	3.1	1.0	2.21	3.42	4.63	5.84
	0.7	11.6	5.8	2.2	0.72	”	”	”	”
	1.0	7.0	3.5	1.4	0.45	”	”	”	”
1200	0.4	12.0	6.0	2.3	0.75	2.36	3.72	5.08	6.52
	0.7	8.6	4.3	1.6	0.55	”	”	”	”
	1.0	5.2	2.6	1.0	0.32	”	”	”	”
1000	0.4	8.1	4.1	1.6	0.52	2.54	4.08	5.62	7.16
	0.7	5.8	2.9	1.1	0.37	”	”	”	”
	1.0	3.5	1.8	0.7	0.22	”	”	”	”
800	0.4	5.6	2.8	1.1	0.35	2.78	4.56	6.34	8.12
	0.7	4.0	2.0	0.75	0.25	”	”	”	”
	1.0	2.4	1.2	0.45	0.15	”	”	”	”

In the heating of some materials, underheating or overheating is of little importance. But, for other materials, ±25 F make a great difference in the properties of the materials. Obviously, a high excess temperature of the furnace gases makes a great difference in the speed of heating and in the first cost of the furnace. For that reason, a calculation of the temperature difference along the furnace is desirable. At this point it may well be mentioned that the temperature difference between the incoming and outgoing ends is small, if the gas travel is short. Excessively long travel is avoided by admitting the hot gases in the center and taking them off at the ends, or vice versa (see Fig. 10). The method of accounting for the time lag and the lower temperature at the outgoing end is explained in the following example.

EXAMPLE. Steel disks 24 in. diameter and 0.20 in. thick, weighing 25 lb each, are to be heated from 100 F to 1050 F in a forced-convection-type furnace 5 ft wide by 10 ft deep, with hot gases at 1100 F. Production is to be 1500 lb per hr. The furnace is charged with 10 disks at a time. What circulation of hot gases is required?

Time in Furnace. $t = \dfrac{10 \text{ disks} \times 25 \text{ lb}}{1500 \text{ lb/hr}} = 0.167 \text{ hr}$

Area of Steel in Furnace. $A = 10 \text{ disks} \times 6.3$ sq ft (both sides) $= 63$ sq ft

Weight in Furnace. 10 disks × 25 lb = 250 lb

Specific Heat of Steel at These Temperatures. $c_p = 0.135$; T_i, T_f, T_g = 100 F, 1050 F, and 1100 F, respectively.

The required overall coefficient of heat transfer may be determined in two ways. One method is by the use of the logarithmic equation 9 which is based on a constant coefficient of heat transfer, and is fairly accurate when the percentage of heat transferred by convection is very high. By this equation

$$k = \frac{250 \times 0.135}{63 \times 0.167} \times \ln \frac{1100 - 100}{1100 - 1050} = 9.55 \text{ Btu per (sq ft, hr, F)}$$

The active temperature difference in this equation is the logarithmic mean temperature difference, which for the above value, is

$$\frac{(1100 - 100) - (1100 - 1050)}{\ln \dfrac{1100 - 100}{1100 - 1050}} = 317 \text{ F}$$

This difference corresponds to 72 per cent of the temperature rise of the stock, which is correct for convection heating only.

The other method employs the "two-thirds" rule, which means that the mean temperature of the stock is two-thirds of the total rise. The rule is based on a variable coefficient of heat transfer by both convection and radiation. By this rule, the equation becomes

$$k = \frac{250 \times 0.135}{63 \times 0.167} \times \frac{(1050 - 100)}{\frac{1}{3} \times 950 + 50} = 8.2 \text{ Btu per (sq ft, hr, F)}$$

From Table 9, where for the present example the value of R is about 0.4, the emissivity of the steel is 0.4, and T_g is 1100 F, the coefficient of heat transfer by radiation only is about 5.0. The convection component must be 8.2 (by the two-thirds rule) − 5.0 = 3.2 Btu per (sq ft, hr, F).

From the convection equation 3 in Chapter 2,

$$3.2 = 1 + 2.7 \times 0.08 \times \frac{520}{1560} \times c$$

and c (velocity of gases in ft/sec) = 30.8.

If the circulation is arranged from one end of the furnace to the other end, and if baffles are arranged for 10 sq ft of cross-sectional area, the circulation of hot gases by the fan will be 10 sq ft × 30.8 ft/sec × 60 sec = 18,480 cfm.

The temperature of the disks at the cooler end of the furnace depends on the method of operation of the furnace. If the furnace is hot, when the disks are charged and the disks at the cool end are charged first, there can be only an insignificant difference of temperature between the disks at the two ends of the furnace, especially so if the disks at the hotter end are removed first. In another method of operation, the temperature of the gas can be reduced to 1050 F as soon as the disks at the hot end have reached that temperature. And if the disks are charged into a cold furnace, a much higher gas velocity is needed to heat the disks in the specified time.

If the disks could be charged simultaneously and could be removed simultaneously, there would be a temperature difference between the disks at the two ends. No analytical method exists for calculating this difference. The following method furnishes an approximation. While the gases pass through the furnace, they lose temperature; much at first, and then less and less. By transfer of heat to the charge (at average temperature of the charge) the drop of temperature of the gas is found from the following equation:

$$\frac{8.2 \text{ Btu (sq ft, hr, F)} \times 63 \text{ sq ft} \times 367 \text{ F temperature difference}}{18{,}480 \text{ cfm} \times 60 \text{ min} \times 0.0267 \text{ lb/cu ft} \times 0.25 \text{ specific heat}} = 25.5 \text{ F}$$

The lower temperature of the heating medium would cause the same lowering of the temperature of the disks, if it were not for radiation from the initially hot walls. It is estimated that this radiation reduces the temperature drop by about 5 F, which means that the disks at the cooler end have reached 1030 F, when the disks at the hotter end have been heated to 1050 F.

Determination of the effective area A of the heating surface is usually more difficult than it is in the example of heating disks. In a furnace such as illustrated by Fig. 61, the extent of the surface contacted by the gases is uncertain. The contacted heating surface and also the free cross-sectional area vary with the shape of the objects in the charge. For that reason the velocity of the gases is usually referred to the empty furnace and is then multiplied by a factor of experience.

Mention is here made of the fact that, in the heating of aluminum, the excess of temperature of the heating gas over the final temperature of the charge may safely be greater than in the heating of steel, because of the low emissivity and the high thermal conductivity of aluminum.

The difference (gas temperature–final temperature of stock) must be small if the property of the heated material is adversely affected by slight overheating, and if the heating stock has low thermal conductivity. This combination is encountered in the annealing of thick coils of thin strip steel. Although the strip is coiled under tension, successive wraps have contact in spots only, because the apparently smooth surface of the strip has microscopically small hills and dales. The latter are filled with air, which, if confined, is a very poor conductor of heat. Since the thickness of the films of air is not known, the time that is required for heating coils is largely a matter of experience. A furnace for heating coils is illustrated by Fig. 62. The following data were derived from experience with such furnaces.

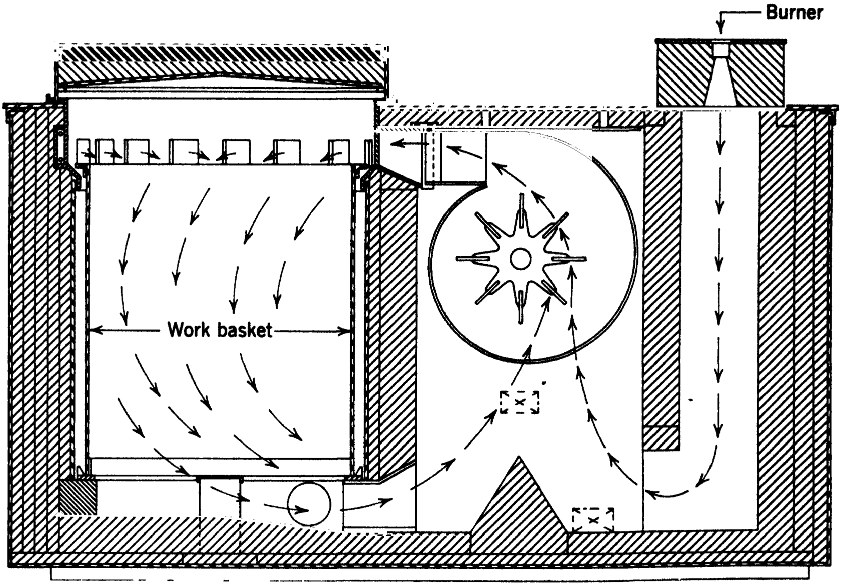

Fig. 61. Gas-fired tempering furnace. (Courtesy of Lindberg Engineering Co.)

With practically all of the heat entering radially from the outer and inner surfaces of the coil, the heating time per inch of radial thickness of auto-body strip is 2 hr for commercial quality and 2.8 hr for deep drawing quality. Both figures include soaking time. In comparison with the forge-plant rule that heat penetrates ⅛ in. in 5 min, the figures given above are equivalent to ⅛ in. in 30 min and in 42 min, respectively.

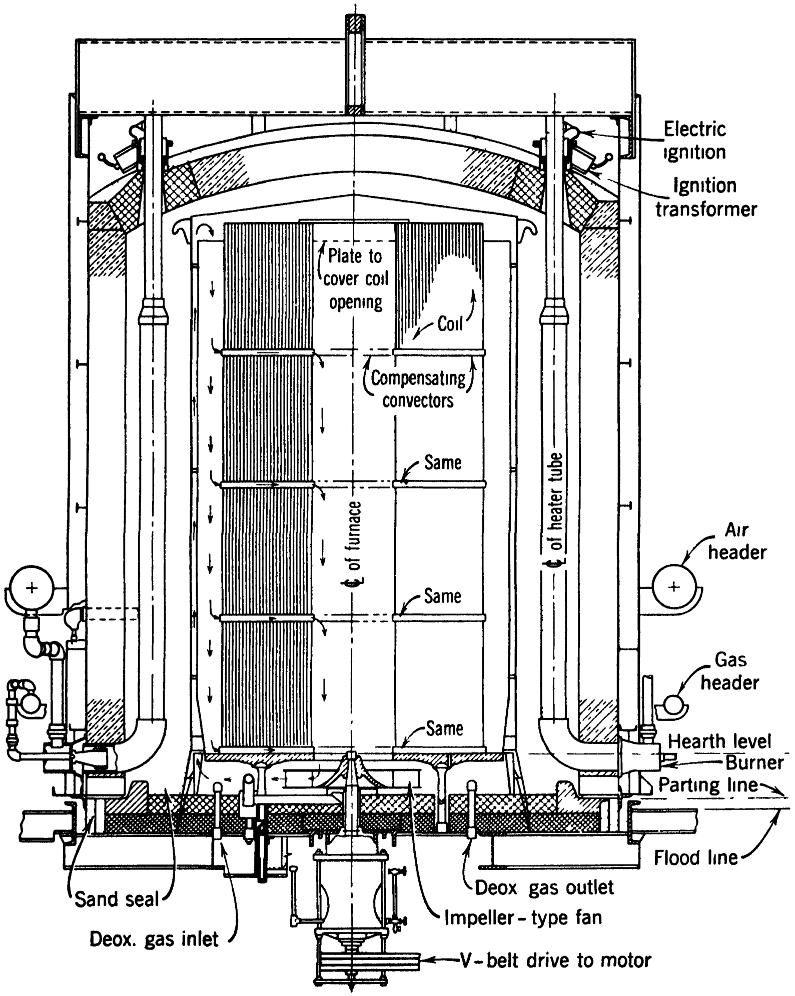

Fig. 62. Heating of coils by radial and axial input of heat.

The goal is to heat the steel uniformly to about 1250 F. In commercial quality, temperature differences up to 70 F are permitted, whereas, in deep drawing quality, about 34 F is the limit.

The time needed for cooling a coil (under the inner cover) down to a temperature at which it may be exposed to air is approximately 50 per cent longer than the heating time by natural circulation, and somewhat less when water or forced air is used on the outside of the cover. For the sake of economical operation, coils became larger and thicker so that heavy coils had to stay under the hood for a whole week.

The time for heating and cooling was reduced by imparting most of the heat axially, instead of radially as shown in Fig. 62. Convector rings were placed between successive coils in the stack. The hot gas was forced to circulate through these hollow rings, transmitting heat to the rings and, through them, to the ends of the coils. The resistance of the small passages in the convector rings called for more powerful blowers. The power of the blowers is converted into heat. One engineer remarked: "We no longer anneal by Btu, but by horsepower." The heating and cooling periods are indeed shortened. Referring to heat flow from both ends of a coil towards the middle, heat travels ⅛ in. in 9 to 9½ min for commercial quality, and 12 to 13 min for deep drawing quality. These values were derived from numerous tests made by Lee Wilson Engineering Company. Of course, heat travels at the same rate in either case. As above, the difference in time is caused by greater temperature uniformity needed for deep-drawing steels. In a strip of 36 in. width the heat travels 18 in. $= 18 \times 8 \times \frac{1}{8}$ in. from either end. For an average rate of 12 min per ⅛ in., the needed time equals $18 \times 8 \times 12 = 1728$ min $= 28.6$ hr. The thickness of the strip has very little effect on the heating time, although some heat flows radially into the coil. It is evident that the heating time becomes long for wide strip.

The above given figures are for design engineers. Owners and operators think and speak in tons per hour.

The increase in weight of coils (diameter, width) led to schemes for shortening heating-plus-cooling time. One invention is based on replacing the air films (between successive wraps) by films of helium and hydrogen. Another invention is based on loosening or expanding the coils so that the gas can be forced axially through the coils for heating as well as for cooling.

11. FURNACES WITH LIQUID BATHS

The various purposes for which salt baths are used are discussed in Volume II.

In salt baths and metal baths, heat is transferred to the charge mainly by conduction. Convection and radiation play minor roles in furnaces of the bath type. Heat is transferred to the work blanks so rapidly that this function normally does not determine the heating capacity. Two other features determine that capacity. The volume of the bath must be so big that immersion of the work blank reduces the temperature of the bath by only a few degrees. Furthermore, the surface through which heat is imparted to the bath must be large enough to permit heat transfer without damage to the container. If the heat is transferred through a metallic kettle, the kettle area through which heat is transferred must be large enough to prevent injury to the kettle by oxidation or overheating. If heat is transferred to the bath from above by radiation and convection, the heat-receiving horizontal surface of the bath must be large enough to prevent injury to the refractories above the combustion space.

The heat to be imparted to the charge during one immersion equals weight times temperature rise times specific heat. The word "charge" includes containers for work blanks. Additional heat input is needed to make up for the heat that is lost from the bath by radiation from the uncovered surface of the bath. The emissivity of a salt bath is approximately 90 per cent. Lead baths are covered with lead oxide and with charcoal to reduce radiation, depending upon the thickness of the cover material.

Zinc baths for galvanizing purposes lie in steel kettles. The latter are quickly corroded by the zinc, if the wall temperature of the kettle is too high. The corrosion is very costly because of a breakout. The danger point lies at a heat transfer of 14,000 Btu per (sq ft, hr) to the wall of the container. Designers aim at 10,000 Btu per (sq ft, hr) hoping that, at the hottest spot, the rate of heat transfer will not exceed the danger point.

Many salt baths are heated by electrical energy. Resistors are arranged between the kettle and the surrounding furnace wall, or else electrodes reach into the bath from above. Furnaces of this type are discussed in Volume II.

12. EXAMPLES

EXAMPLE 1. FURNACE HEATING SKELP FOR BUTT-WELDED TUBES. The arrangement is shown in Fig. 341. The hearth is 7 ft 6 in. wide by 25 ft long. Average height of arch above hearth is 4 ft 0 in. Skelp 11 in. wide by 0.22 in. thick by 23 ft long is to be heated to 2550 F, the furnace temperature being 2600 F. Find the heating time per piece, the heating capacity of the furnace and the temperature difference in the material.

The stock lies flat on the hearth, which is practically covered; hence we must figure on heat entering from one side only. The flame ranges from semi-luminous to luminous, depending partly on the quality of the fuel gas.

(*a*) Heating time by method of page 60. Average temperature difference between furnace and stock = ⅓(2550 − 100) + 50 = 866 F. From Fig. 52 for $R = 0.25$ and stock emissivity of 0.9, k is about 35 Btu per (sq ft, hr, F). Adding 10 per cent for luminosity, the figure becomes 38.5. Heat input required (Fig. 27) = 408 − 11 = 397 Btu per lb of steel. Weight of material per sq ft of surface exposed to heat = 0.22 × 144 × 0.283 pci = 9.0 lb. Heating time

$$= \frac{9\text{ psf} \times 397\text{ Btu per lb}}{38.5\text{ Btu/(sq ft, hr, F)} \times 866\text{ F temp. difference}}$$

$$= 0.107\text{ hr} = 6\tfrac{1}{2}\text{ minutes}$$

(*b*) Capacity of furnace

$$= \frac{9.0\text{ psf}}{0.107\text{ hr}} = 84\text{ lb per (hr, sq ft of hearth actually covered)}$$

Either 6 or 7 pieces could be on the hearth at a time (11 in. width of pieces, 90 in. width of hearth). If 7 pieces, then the heating capacity of the furnace is

$$\frac{7 \times 11/12 \times 23\text{ ft} \times 9\text{ psf}}{0.107\text{ hr}} = 12{,}400\text{ lb per hr}$$

or 66.0 lb per (hr, sq ft of total hearth area).

(*c*) Uniformity of temperature in the stock. By the rule that heat travels ⅛ in. in 5 min, the skelp should remain in the furnace 5 × 2.2/1.25 = 8¾ min. Since this rule is based on a very small temperature difference in the stock, a heating time of 6½ min is acceptable. Temperature uniformity is made better and heating time is shortened by the fact that the skelp is drawn slowly out of the furnace and through a ring die. During that time the hearth is exposed. The replacement skelp is not entered until the hearth under the preceding skelp is entirely bare.

There is some space between adjacent pieces, so that the edges are hotter than the rest of the skelp. The heating process is wasteful, because only the edges must reach welding heat, when the rest of the skelp could be a dull red heat.

EXAMPLE 2. FORGE FURNACE FOR HEATING LARGE ROUNDS. The arrangement and dimensions of the furnace are shown in Fig. 63. Let it be required to heat forging ingots of mild steel, 24 in. in diameter, from 100 to 2200 F. Find the time required for heating and the minimum furnace dimensions.

The diameter of the ingot is so great that the heating time is determined by temperature difference in the stock. If the standard method of heating is followed, about 18 min per in. of diameter are needed. If an accelerated method is followed, 14 min per in. of diameter suffice. Depending on the method of heating, the required time ranges between 14 × 24 = 336 min (5¾ hr) and 432 min (7¼ hr). These heating times are based on the ingots being blocked up so that the flame can circulate around it. If the ingot lies on the hearth, the heat has to travel a great distance, and 30 to 40 per cent must be added to the above calculated heating times.

In order to be heated in 6 hr, the ingot must be surrounded by a thick layer of clear products of combustion or a thinner layer of luminous gases having a high temperature (2500 F). The following calculations afford an insight into these relations.

The ingot weighs 245 psf of ingot surface. The heat to be imparted to each pound of the ingot equals 370 Btu. Then the heat per sq ft of surface = 91,000 Btu. If the ingot is to be heated in 6 hr, the hourly (average) input of heat equals 91,000/6, which roughtly equals 15,000 Btu per (sq ft, hr).

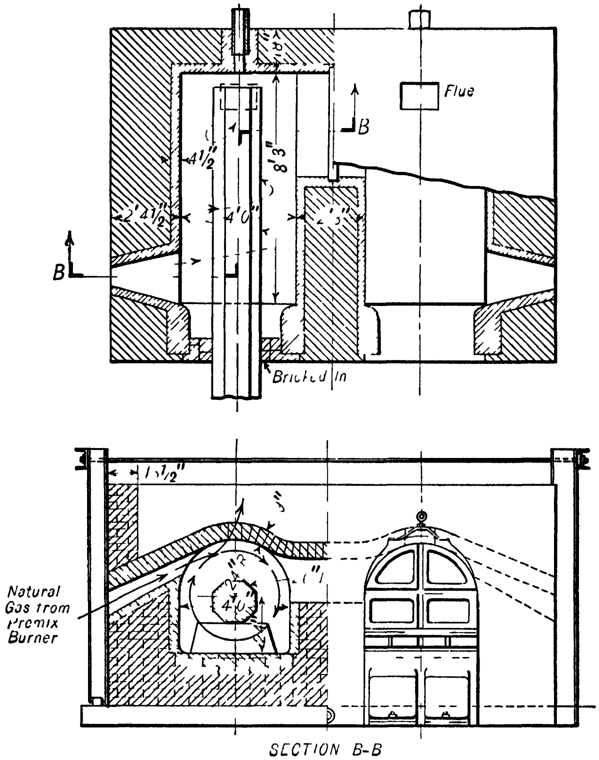

Fig. 63. Furnace for heating forging ingots.

By following the method of Fig. 51, the active mean temperature difference is found to be 700 + 300 = 1000 F. Then a heat-transfer coefficient of 15,000/1000 = 15 Btu per (sq ft, hr, F) suffices. In order to determine the thickness of the gas layer that is needed for producing this heat transfer coefficient, turn to Figs. 41 and 42. If natural gas is the fuel, the products of combustion contain roughly 10 per cent CO_2 and 19 per cent H_2O. Thus, in 1 ft thickness of products of combustion, $p \cdot s = 0.1$ ft atm for CO_2 and 0.19 ft atm for H_2O. The mean temperature between the gases of 2500 F and ingot surface temperature 1500 F equals 2000 F. At that mean temperature, k for $CO_2 = 5.3$ at 0.10 ft atm; and for H_2O at 0.19 ft atm, $k = 7.2$ Btu per (sq ft, hr, F). The sum of the two coefficients equals about 12.5. Hence, a gas thickness of 1 ft suffices.

A few remarks are in order. The heat transfer coefficient is reduced and increased by opposing influences. It is reduced about 15 per cent by the overlapping of radiation bands and by the absorptivity of the surface of the ingot. It is increased considerably more than 15 per cent by radiation from the products of combustion to the furnace walls which re-radiate to the ingot. Although the thickness of the gas layer may be less than 1 ft, this minimum dimension is preferred to reduce the danger of flame impingement. (See also Tables 5 and 6.)

Example 3. Furnace to Heat Billets for Rolling. A gas fired furnace is 35 ft wide by 13 ft deep, with 6 charging doors along the front of the furnace, as shown in Fig. 64. The average height of the roof above the hearth is 4 ft. A total of 40 billets of alloy steel are charged into the furnace when it is at 1500 F, the furnace having cooled to this point while charging the billets. The billets are 5 in. by 5 in. by 10 ft long, and are to be heated from 100 to 2100 F with maximum final temperature difference of 50 F. Determine the heating time.

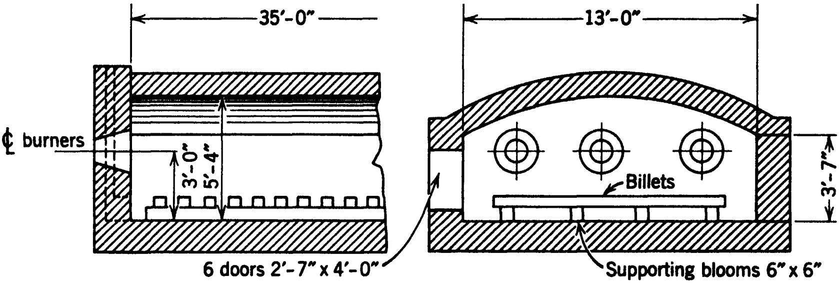

Fig. 64. Batch-type billet-heating furnace showing arrangement of billets in furnace.

The average thickness of the gas layer above the billets is about 41 in. and the layer below the billets is only 6 in. Also note that the top gases are hotter than those below. For both reasons it is judged that five-sixths of the heat enters the top, the remaining one-sixth entering from the bottom. With the billets 5 in. apart in the furnace, it is estimated that the radiation which the vertical surfaces of the billets receive is equivalent to 60 per cent of the full radiation. Hence, 3 in. per side are used in the calculation. Then, the heat absorbing surface per ft of billet length is

$$\frac{5\text{ in. (top)} + 6\text{ in. (two sides)} \times 1\text{ ft}}{12} = 0.92\text{ sq ft}$$

The weight per ft of billet length heated from the top is $\frac{5}{6} \times 85$ lb per ft = 71 lb/ft. The heat to be imparted per lb is 360 − 12 = 348 Btu.

In computing the ratio R, allowing for the fact that the bottom is not as effective as the top of the furnace, the total area of the billets divided by an estimated effective refractory area of 1200 sq ft is 0.56.

With stock emissivity equal to 0.8, $R = 0.56$, and final temperature 2100 F, k from Fig. 52 = 0.86 × 21.7 = 18.6 Btu per (sq ft, hr, F). The average furnace temperature during the cycle is about 1950 F and the average temperature of the billets is 100 + ⅔(2100 − 100) = 1435, so that the mean temperature difference is 515 F.

$$\text{Heating time} = \frac{71\ \text{lb/ft} \times 348\ \text{Btu per lb}}{\underset{k}{18.6} \times \underset{\text{area}}{0.92} \times \underset{\substack{\text{temp.}\\ \text{diff.}}}{515}} = 2.8\ \text{hr}$$

The rate of heat absorption in this case is 18.6 Btu per (sq ft, hr, F) × 515 F = 9600 Btu per (sq ft, hr), which is the same as that found on curve 6 of Fig. 59 for the value of $R = 0.56$.

chapter **4**

The Heating Capacity of Continuous Furnaces

1. CONTINUOUS FURNACES AND RELATION TO BATCH-TYPE FURNACES

In continuous furnaces the stock or charge moves continuously, or intermittently, through them. The stock may be pushed or rolled through, or it may rest on a rotating hearth, or be suspended from an overhead conveyor. If the charge rests on trays that are pushed through, or if it rests on a moving chain or on rollers, continuous furnaces are commonly called automatic.

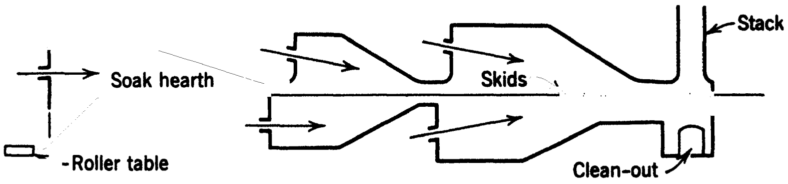

Fig. 65. Continuous furnace, endfired in several zones.

A continuous furnace may be so heated that the temperature of the products of combustion is practically constant everywhere in the furnace. This temperature uniformity may be obtained by front firing in several zones, as illustrated by Fig. 65, or by roof firing, or by side firing in several zones, as indicated in Fig. 66. In such furnaces the heating capacity at least equals and often exceeds the heating capacity of batch-type furnaces. Originally, continuous furnaces were built with a decided temperature drop from the (stock) discharge end to the (stock) entrance end, but since about 1948 continuous furnaces for heating slabs have been built mainly for constant or almost constant temperature in the heating zone, to keep furnaces short. The statements relating to batch-type and continuous furnaces are correct for top-fired furnaces carrying a temperature corresponding to the temperature of the batch-type. The heating capacity of such furnaces is, of course, determined by hearth area, furnace temperature (which is intermediate between gas temperature and wall temperature), emissivity of the stock, composition and thickness of layer of products of combustion, and ratio of wall surface to stock surface.

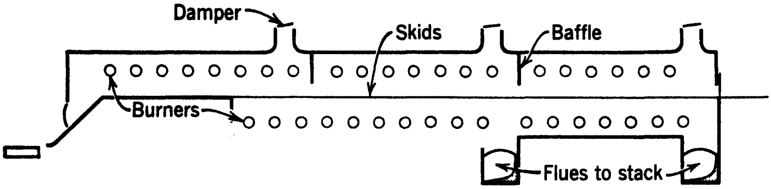

Fig. 66. Continuous furnace fired from the sides.

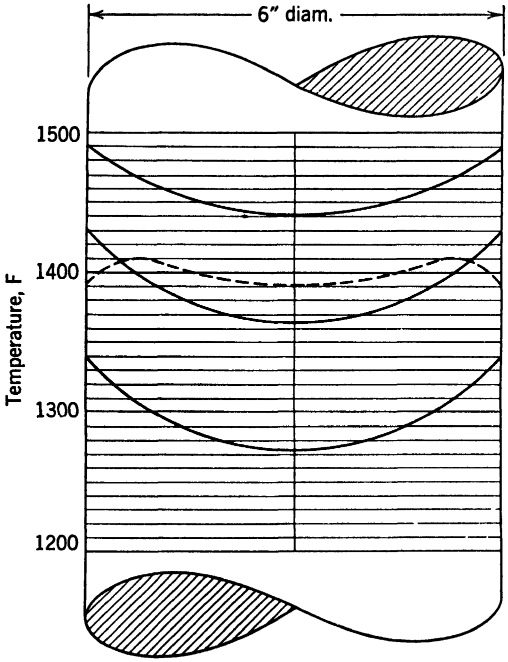

Fig. 67. Temperature equalization obtained by interruption of heat supply.

The heating capacity of continuous furnaces can (and usually does) exceed that of batch-type furnaces having the same hearth areas, for two reasons: First, whereas, in batch-type furnaces (with few exceptions, explained in Chapter 3) temperature must be held down to prevent overheating, the temperature in the heating zone of a continuous furnace may be very high, if thin stock is removed promptly on having reached the desired temperature or if, in the heating of thick stock, the furnace is equipped with a soaking zone in which the temperatures can be equalized as shown by the dotted curve in Fig. 67. If the stock has a high thermal conductivity, the soaking zone is not needed. The second reason is that in a pusher-furnace the heating stock may be supported by skid pipes so that much heat can be imparted to the bottom of the stock. Both features are discussed below.

2. CONTINUOUS FURNACES FOR HIGH TEMPERATURES

Thickness of heating stock does not limit heating capacity as much in continuous furnaces as it does in furnaces of the batch type, because heat can be imparted to the stock from below. The limiting effect of thickness varies with the thermal conductivity of the stock and with the required uniformity of temperature in the heated material. Figure 68 contains the relations between thickness of stock, heating time, temperature difference in stock, and rate of heating in lb/(sq ft, hr) for mild steel that is heated to 2200 F.

"Triple" firing of continuous furnaces refers to top heat, bottom heat, and separate firing of the soaking zone (Fig. 69). In comparing heating capacities of such furnaces, statements regarding the hearth area of reference should be specific: whether top heating zone only, or top plus bottom area, or top plus bottom plus soaking zone, and finally whether surface of charge or hearth area is referred to.

(a) Factors Limiting Heating Capacity. In calculating heating capacity of continuous furnaces that are equipped with soaking zones, there should, ideally, be no transfer of heat in such zones. In fact, a slight loss of heat from the top would speed equalization of temperature. This method is used in top-fired furnaces that heat long square billets for press piercing.

In furnaces that are equipped with skid pipes, the soaking zone serves mainly for elimination of black spots, as explained below.

Example. Two-inch steel billets are pushed through a continuous furnace in which a constant gas temperature of 2400 F is maintained. For the conditions stated in Fig. 50, the heating time to 2200 F is 1 hr. One square foot of heating stock weighs roughly 80 lb. Hence the billets are heated at the

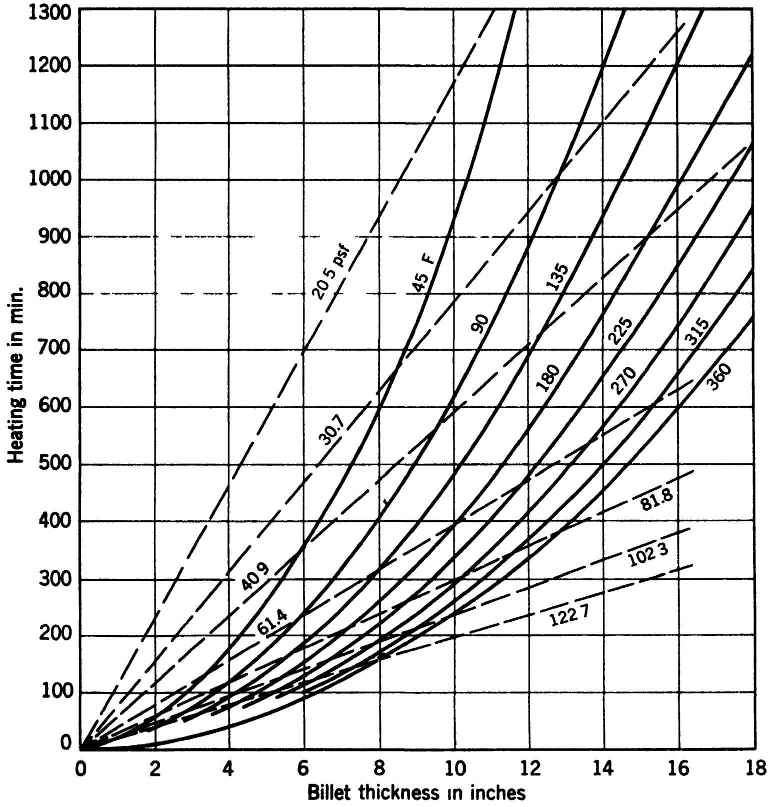

Fig. 68. Temperature differences in steel billets as a function of billet thickness and heating time. Top flame only. For equal inputs of heat from above and below figure with half the actual billet thickness.

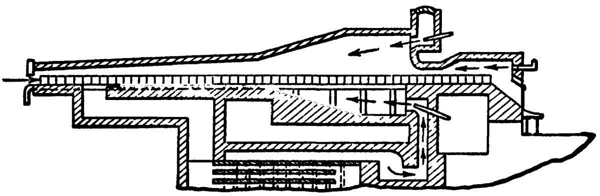

Fig. 69. Triple-fired zone heating furnace. (Courtesy of Rust Engineering Company.)

rate of 80 lb per (sq ft, hr). If only 85 per cent of the hearth is covered by steel, the rate per (sq ft, hr) of steel-covered surface remains approximately the same, but the rate per square foot of hearth becomes 0.85 × 80 =

68 lbs per (sq ft, hr). The stock is so thin that supply of heat from below offers no advantages.

Figure 50, previously referred to, teaches the following: When flame temperature exceeds final temperature of stock by a small amount only, the time-temperature curve is almost parabolic. But, if flame temperature greatly exceeds the final temperature of the stock, the time-temperature curve is almost a straight line. The result is that, in the heating zone of modern continuous furnaces, the value ½ should be used for the mean active temperature difference, replacing the ⅓ used in the calculation of batch furnaces.

Increase of output may be obtained by burning more fuel or by preheating the combustion air, whereby average flame temperature and furnace temperature are increased. Again, by reference to Fig. 50 it is found that the 2-in. plate reaches a surface temperature of 2200 F in ½ hr. This heating time corresponds to a heating rate of $2 \times 80 = 160$ lb per (sq ft, hr) of steel-covered hearth, or to a rate of $0.85 \times 160 = 136$ lb per (sq ft of total hearth, hr).

This high furnace-capacity is brought about by the great difference of temperature between flame and heating-stock and by the greatly increased coefficient of heat transfer, which is a function of flame temperature. In Fig. 52, where the solid curves pertain to batch-type furnaces, heat-transfer coefficients for higher excess flame temperatures than are permissible in batch furnaces are shown by dotted curves. If, for instance, the stock is to be heated to 1800 F, then an excess temperature of 15 per cent means a flame temperature of $1.15 \times 1800 = 2070$ F.

In the heating of 2-in. billets, rates of 130 lb per (sq ft, hr) have been attained with front-firing, top heat only. Furnace operation at high rates requires alertness of the crew, quick action in case of delays in the processing equipment, and excellent maintenance.

So far as uniformity of temperature is concerned, heating rates in excess of 120 lb per (sq ft, hr) are permissible for 2-in. billets. From Fig. 68 it follows that, in the example just given, the bottom of the billets would be about 90 F cooler than the top. If 8-in. steel billets were heated at the same rate, their tops would melt together, while the bottom would be still cold. Even at the rate of 80 lb per (sq ft, hr), the bottoms of the blooms would be 200 F colder than the tops (see Fig. 68). This temperature difference is too great for most of the subsequent steps in processing. Either the rate of heating must be reduced, or bottom heat must be supplied.

If only three-eighths of the total heat input comes up from the bottom. the two heat flows meet at 5 in. from the top, where (Fig. 68) the temperature is 150 F lower than on the surface. If greatest possible heating capacity in a given space is desired or necessary, the temperature in the heating zone is run up as high as circumstances permit (explained below) and some equalization of temperature, including elimination of black spots, is obtained in a soaking zone. The length of the soaking hearth is determined by temperature difference between surface and core (in very thick sections) and by elimination of black spots (in medium heavy sections). In the rolling of thin strip sheets, micrometer measurements in the finished product reveal the location of the black spots in the slab. For that reason, the length of the soaking zone depends upon the stringency of specifications on uniformity of thickness in the finished material. Expressed differently: The capacity of a furnace with a given length of soaking zone depends on the required uniformity of gage in the finished product. This fact explains the seemingly illogical practice of adding top heat in the soaking zone. Elimination of black spots is considered to be more important than top-to-bottom uniformity of temperature.

In average practice, the aforesaid rigid specifications do not apply. In conformity with varying requirements, the length of the soaking zone ranges between one-eighth and one-third of the length of the high-temperature heating zone. Rotating hearths have no water-cooled skid pipes. In consequence, the soaking zone can be shorter; it usually is less than one-eighth of the length of the heating zone. Very rapid heating results in a short heating zone, but requires a long soaking zone for thick material of low thermal conductivity.

In the heating of steel, the rate of heating is seldom, if ever, limited by the strength of the refractory materials, except in a few unusual designs.

(b) Front-Fired Continuous Furnaces. For greatest uniformity of temperature, top- and bottom-fired continuous furnaces with almost constant temperature from end to end, plus a soaking zone, represent the ultimate heating capacity per unit of hearth area and in unit time. However, the greatest number of continuous furnaces for forging or rolling are, at the time of this writing, operated with a drop of temperature and with front firing only. The heating capacity of such furnaces is limited by several circumstances. The average temperature difference between products of combustion and stock is lower than in a furnace in which the temperature is uniformly high. Front firing has the disadvantage that the rate of heat transfer is greatest at the place where the surface temperature of the stock is highest. This statement is illustrated by Fig. 70.* The dotted lines show the temperature of the products of combustion for different rates of firing. The solid lines show the rise of the temperature of the stock passing through the furnace. The rate of heating in pounds of steel per (sq ft, hr) is marked on both sets of temperature curves. At the heating rates commonly used in front-fired continuous furnaces without recuperation, 50 to 70 lb per (sq ft, hr), the temperature of the stock does, indeed, rise most rapidly at the hot end.

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* The method by which Fig. 70 was calculated is explained in Chapter 5.

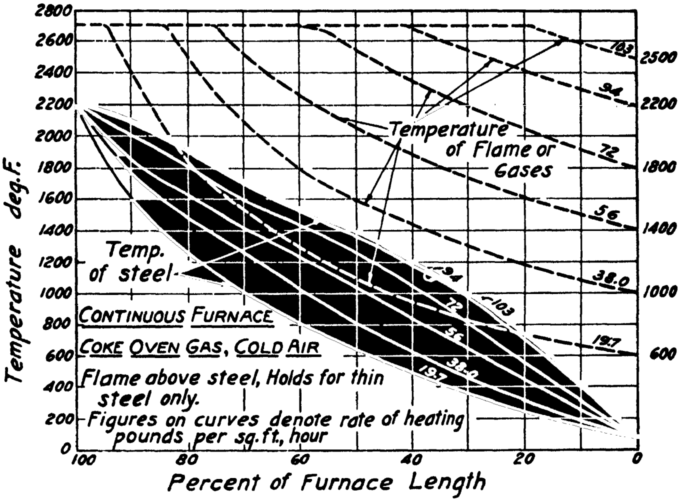


Fig. 70. Temperature variation of steel and of gases in a continuous furnace heated by coke-oven gas.

One result is that there is little time left for temperature equilization in furnaces without soaking zones. The higher the rate of heating, the greater the temperature differences in the stock, unless the stock has a high conductivity and a low emissivity. Another result is the melting of corners and edges of the charge and the melting of scale. The mixture builds up on a solid hearth or drops down into the space under the skid rails, where it freezes and makes a cake that must be removed over the week-end.

These troubles are avoided if, in the heating of steel, the furnace temperature at the hot end is kept below 2350 F. In that case, the following heating capacities are found for pusher-type furnaces with top heat: Up to 60 lb per (sq ft of hearth, hr) when heating 9-in. blooms. If very heavy billets such as 18-in. by 18-in. are heated, 40 lb per (sq ft, hr) are the limit.

At this point the properties and effects of scale may well be discussed.

Under average conditions, the weight of scale on the surface of steel can be expressed by the equation

$$\text{lb of scale per sq ft} = 0.4 \times (T/2200)^5 \times t \qquad (11)$$

where T equals furnace temperature degrees Fahrenheit, not absolute, and t is time of exposure, hours. The accuracy of this equation is about ±25 per cent; composition of steel and of furnace gases and method of circulation of gases have great effect on scale formation. Equation 11 is purely empirical, based on observation, and has no known theoretical foundation.

Steel scale begins to soften at 2200 to 2300 F, depending on its composition. It melts at 2300 to 2400 F, again depending on its composition.

From the last two paragraphs it follows that, if thick steel (which stays in the furnace for a long time) is heated in a hot furnace, the scale becomes mushy, if not liquid.

Semi-molten scale has caused many erroneous temperature measurements in steel-heating furnaces. Scale is an insulator. If the scale is not shiny or glossy, optical pyrometers and radiation pyrometers measure scale temperature, but not steel temperature. And, if the scale is shiny, the pyrometers indicate a temperature somewhere between roof temperature and scale temperature.

(c) Front Firing, Top and Bottom. The heating capacity of furnaces with top and bottom heat is less than twice those in (b) because a soaking zone must be provided for the elimination of the black spots that were caused by the water-cooled skid pipes. The maximum rate is about 95 lb per (sq ft, hr), again with 2350 F furnace temperature and proportionally less for very heavy stock. These values can be increased by alternating short-flame and long-flame burners. The latter carry the high temperature farther back into the furnace. So-called high-velocity burners have a similar effect. Of course, the heating rates can be increased by higher furnace temperature (resulting in greater operating difficulties).

(d) Side Firing. The heating capacity of continuous furnaces is greatly increased by adding midway firing or side firing (or a combination of both) to end firing, as schematically illustrated by Figs. 65 and 66. By these means, capacity can be increased about 50 per cent. At the time of this writing, slab-heating furnaces for strip mills are rated at 150 lb per (sq ft, hr) on low-carbon steels. It may be repeated that some properties of the heating stock reduce capacity.

Among them are: Shiny surface, low thermal conductivity at low temperature, brittleness at low temperature, and thickness.

Furnaces with rotating hearths have no ends and, therefore, cannot be endfired, although they can be fired through dips in the roof. Such furnaces are sidefired. Their heating capacity is about 25 per cent higher than that of top-fired and endfired furnaces of the pusher type.

The above given values are based upon the ratio *R* (heat absorbing surface of stock to surface of furnace-interior) ranging between 0.3 and 0.4.

As previously mentioned, the heating capacity of continuous furnaces varies greatly with the nature and surface condition of the heating stock. High-carbon steel is not heated to as high a temperature as is low-carbon steel, to avoid decarburization, with the result that the heating capacity is lower. Copper is heated to an even lower temperature. (See section 4 of this chapter.)

It is interesting to note that, in furnaces with top and bottom firing, as much as 25 per cent of the total heat input occurs in the soaking zone. The conclusion, as previously stated, is that temperature equalization between surface and interior is considered to be of less importance than elimination of black spots. The soaking zone eliminates black spots, but does not necessarily eliminate cold spots which show up as greater thickness in the finished flat product.

(e) Limitation in Hearth Length by Pushing of Stock. Another factor limits the capacity of pusher-type continuous furnaces with regard to pounds heated per hour, but not with regard to pounds heated per square foot per hour. That factor is: Safe length of hearth. By "safe length" is meant a length that prevents upward buckling and piling up. The safe length depends upon the flatness of the hearth, thickness of the stock, and shape of the contacting surfaces of the stock. Thin billets are seldom straight, and often have sheared ends that are irregular. Very cold bars rise in the middle. A hearth length that is safe in one mill causes buckling in another mill.

If the hearth is horizontal, the pusher force equals weight (W) of stock times friction coefficient (fr). W is proportional to the length of the hearth. The pusher force for unit width of stock is proportional to length of hearth times thickness of stock. Although the equation for buckling of columns does not exactly apply in this case, it gives an approximate idea of the relation between thickness of stock and safe length of hearth. From Euler's equation it may be concluded that the safe length of hearth is proportional to the two-thirds power of the thickness of the stock, if the hearth is horizontal. For a stock four times as thick, the safe hearth may be $2\frac{1}{2}$ times as long.

Inclining the hearth increases the safe length. This is the principal reason why furnaces for heating thin stock have inclined hearths. Hearth inclination reduces pusher force in accordance with the equation

$$\text{Pusher force} = Wfr \cos j - W \sin j \tag{12}$$

where j is the angle between the hearth and the horizontal. If $\tan j$ equals fr (coefficient of friction), the pusher force is reduced to zero.

Excessive hearth inclination interferes with pressure conditions in the furnace, as explained in Chapter 8. An inclination of more than 8 degrees is rare. The safe length of hearth also depends upon the shape of the contacting surfaces of the billets. If the billets or slabs have round edges, climbing occurs easily. Crooked billets likewise favor climbing. Means for reducing the danger of piling up are discussed and described in Volume II.

Round billets for piercing cannot safely be pushed through a furnace; the inclination of the hearth is so selected that newly charged cold rounds will almost roll down by gravity. Tubes present the same problem.

Hearths up to 100 feet long are in use for heating thick stock. For billets 2 in. thick, hearth lengths up to about 50 ft are in use. For drop forging, shorter hearths are commonly used.

Plates are not pushed through furnaces, but are heated either in furnaces of the batch-type in continuous furnaces of the rotating hearth, walking beam, or roller hearth type. Plates are usually heated for annealing, bending, flanging, or similar operations, but because they are seldom heated to high temperatures, the continuous-conveyor types are preferred, as illustrated in Volume II. In annealing of plates the capacity of such furnaces is low, ranging between 18 and 25 lb per (sq ft, hr). If the plates are heated for bending or flanging, the furnace temperature is somewhat higher and the heating capacity is greater, up to 50 lb per (sq ft, hr).

(f) Capacity of Continuous Furnaces from Practice. The numerical values for the capacity of steel-heating furnaces are based upon uninterrupted operation throughout the work-week. Delays in the mill or in the forge shop reduce the weight of steel heated in the furnace, but do not reduce the heating capacity of the furnace.

Figure 71 furnishes information on the rates at which heat is transmitted to the exposed steel surface in continuous furnaces. Curves

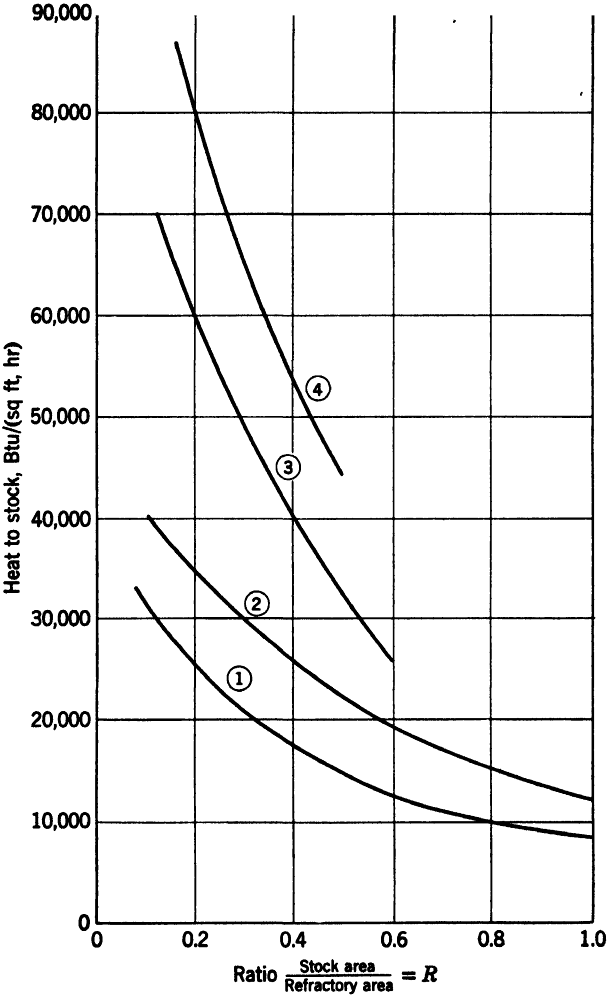


Fig. 71. Overall rate of heat transfer for continuous steel-heating furnaces.

These average values for the rate are taken from practice and apply to the following conditions:

Curve 1. Billet or slab heating from top only, one zone, endfired.
2. Billet or slab heating, top and bottom, three zones.
3. Axial heating, high temperature, at recommended rates, fired full length of furnace.
4. Axial heating, high temperature, ultimate rates, fired full length of furnace.

Thickness of gas blanket $s = 2\frac{1}{2}$ ft. For definition of ratio R, see p. 100.

1 and 2 are for pusher-type furnaces, and curves 3 and 4 are for axial-type continuous furnaces, described in the next section. In each instance, the relations shown are based on the total exposed steel area, including the soaking zone of the furnace. *R* is the ratio of exposed surface of the stock to total refractory area, including the not covered part of the hearth. Thickness of the gas blanket equals about 2½ feet. Ratio *R* is seldom less than 0.3, except when short pieces must be heated in a wide furnace. In that case, the heating capacity of the furnace is not utilized.

As previously mentioned, the rates shown in Fig. 71 may be, and have been, increased by changes from conventional design, or by heavier firing rates, at the expense of increased maintenance.

3. AXIAL CONTINUOUS FURNACES FOR TEMPERATURES ABOVE 2000 F

Certain hot forming processes, such as the piercing of tube rounds, rolling strip in planetary mills, and continuous butt welding of tubes, are facilitated by heating the stock while it travels axially through the furnace. In spite of the moderate speeds of the processing equipment, axial continuous furnaces are long; hence, the desire to shorten them by the greatest possible transfer of heat.

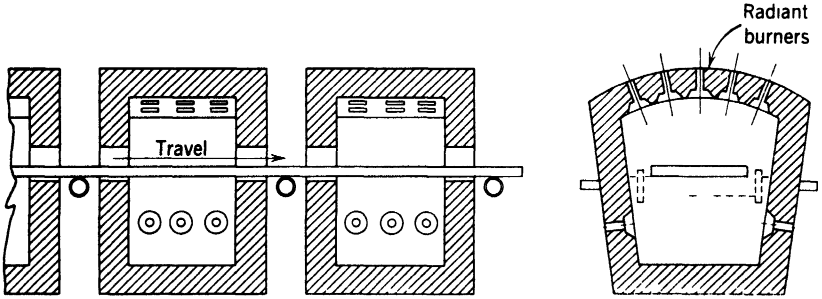


Fig. 72. Axial heating furnace, barrel type.

In the heating of billets, slabs, or strip to high temperatures, furnaces of two kinds are in use. They are diagrammatically illustrated by Figs. 72 and 73. The traveling stock does not rest on the hearth; hence, the term, lb per (sq ft of hearth, hr) is both imaginary and arbitrary. In spite of this fact, the width of the furnace is often considered to be the width of the imaginary hearth.

Furnaces of the type illustrated in Fig. 72 come close to the ideal type for which Table 5, Chapter 2, was calculated, because heat is radiated equally to all parts of a single piece. The transfer coefficients of Table 5 may, therefore, be applied with little or no reduction. The heating time is calculated by the logarithmic equation 9.

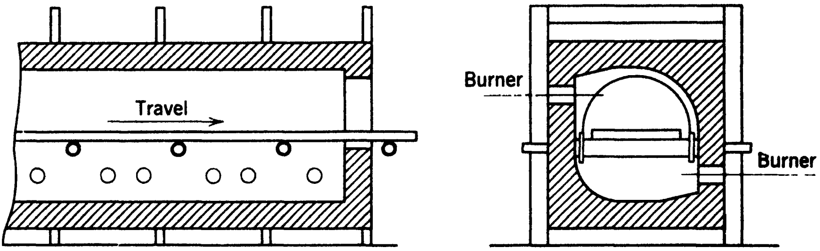


Fig. 73. Axial heating furnace with internal rollers.

The required length of the furnace equals heating time times velocity of stock. The heating time for Fig. 72 is calculated in steps for greatest accuracy because the furnace is built up of individual barrels. The traveling stock is supported by rollers between barrels, away from the heat of the furnace. The barrels must be short enough to prevent sagging of the hot stock, particularly at the leading end of the stock. Between adjacent barrels, the stock loses heat. This fact should be considered in the calculation but, because of the short space between successive barrels, it is seldom considered in practice. Heat is imparted to the stock at a high rate, to shorten the furnace, which in some cases exceeds 100 ft in length. The larger the inside circumference of each barrel in relation to the dimensions of the stock, the higher is the coefficient of heat transfer, and ratio *R* is purposely low in this type of furnace. If the traveling stock is thick, the last zone carries a lower temperature for the sake of temperature equalization within the stock.

In furnaces of the type illustrated in Fig. 73, water-cooled rollers absorb much heat, and for that reason, underfiring must be greater than top firing. In furnaces of the type shown in this figure, slabs 2¼ in. thick are being heated at the rate of 330 lb per (sq ft, hr) of upper surface. If referred to both top and bottom surface, the rate is 165 lb per (sq ft, hr). The products of combustion must be discharged at very high temperatures at intervals along the length of the furnace. The fuel economy is very poor, as explained in the next chapter, and recuperators are frequently used.

If, in axial heating, the stock is a continuous bar or strip, fewer supports are required, because there are no free ends which can bend down and catch on supports that are spaced far apart. The reduced number of supports absorb less heat. Figure 74 illustrates this type of furnace for heating skelp for the continuous butt welding of pipe.

For this process, temperature uniformity in the strip is no object. The purpose is to heat the edges of the strip to white-hot (dripping) temperature to produce a sound weld.

4. CONTINUOUS FURNACES FOR HEAT TREATMENT OF STEEL (1200 TO 1600 F)

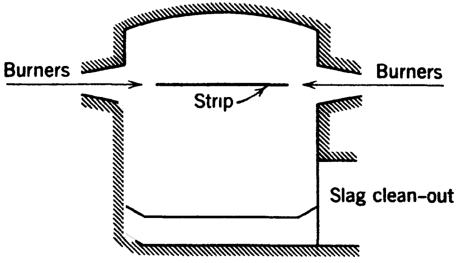
The specific heating capacity of these furnaces is smaller than the heating capacity of furnaces that serve for forging or rolling steel, because the temperatures are lower and the requirements for uniformity of temperature are greater. Uniformity of temperature requires slow heating, particularly of heavy pieces. For that reason, heat-treating furnaces are fired from the sides, usually in several zones for better temperature control than can be obtained with front firing. Pieces in the charge have, in many cases, been rough-machined, with the result that their emissivity is low. Furthermore, metallurgical specifications frequently require time for holding at temperature and for slow cooling. Strictly speaking, these last two periods are not part of the heating capacity of a furnace. If they were, the same furnace would have different heating capacities.

Few pieces to be heat-treated lend themselves to being pushed through a furnace. Large pieces are moved through the furnace by walking beams or rotary hearths, while smaller pieces are usually carried on shoes or trays that are moved by special conveyors, as described and illustrated in Volume II. In the calculation of heating capacity, the weight of trays or conveyors must be taken into consideration.

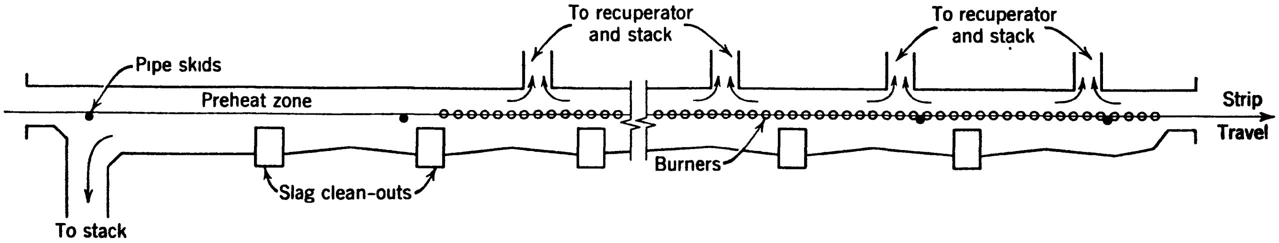
If the charge is to be heated to temperatures of about 1400 F or less, heating by forced convection is preferred, as explained below.

Because heat-treating furnaces are fired from the sides, either above or below the heating stock or both, and because they have an almost constant temperature of products of combustion along the travel of the heating stock, the heating time can be calculated by the methods explained in Chapter 3. Near the entrance of the cold charge the rate of heating may be high, but from the point where the surface of the stock has approached the final temperature, firing must be reduced.

Many heat-treating furnaces are filled with protective gas atmos-



Cross section



Longitudinal diagram

Fig. 74. Diagrammatic arrangement of axial furnace for heating skelp for welded tubing.

pheres. In such furnaces, the source of heat is either electric resistors or radiant tubes. The capacity of electrically heated furnaces is discussed in Volume II. The heating capacity of radiant tubes is limited by the resistance of the tube material to oxidation and creep. Both properties limit the tube temperature and the closeness of tube spacing. Overheating of those parts of the tubes which are adjacent to the wall is avoided if the tubes are spaced at a minimum of two tube diameters apart, and if free circulation around the tubes is provided. This spacing leaves an open space of one tube diameter between the tubes.

If the charge is free from scale, radiant tubes are placed both above and below the charge. As the heating stock travels through the furnace, the stock is exposed to radiation of fluctuating intensity. In calculating heating capacity, the radiating capacity of each tube may be spread uniformly over one space.

The safe temperature of the outside of a radiant tube depends upon both the material of the tube and the method of support. Tube temperatures up to 1800 F are safe with proper alloys, and higher temperatures are possible with more expensive alloys. Tubes of silicon carbide may safely reach a temperature of 2500 F. Spacing of tubes less than two diameters between centers shortens the life of the tubes. Wide spacing reduces heating capacity. This capacity may be calculated, if the permissible temperature of the radiant tubes is known. Data on resistance to oxidation and creeps are given in Chapter 6.

If radiant tubes are located above the charge, heat transfer by convection is small. Convection heat from tubes that lie under the charge is somewhat greater, but is also quite limited because of the small temperature difference between tube and furnace gases. As far as the authors know, no test data on convection under furnace conditions are available. Uniformity of heat transfer in such furnaces is frequently improved by the use of "beater fans" extending through the roof, which insure a uniform movement of the furnace atmosphere to all parts of the furnace.

Continuous furnaces of the axial type, such as catenary furnaces, are used in heat-treating practice. Their capacity is, of course, much lower than the capacity of high-temperature furnaces of the same size. Usually, the strip passing continuously through such a furnace has a very low emissivity.

In the consideration of the heating capacity of such continuous furnaces, this book is concerned only with the thorough heating of the product, and not with the additional time which must be allowed for the different metallurgical changes which account for wide variations in the overall heat transfer rate from as low as 2 Btu per (sq ft, hr, F) to 30 Btu per (sq ft, hr, F) for steel treated for different purposes.

Referring to Fig. 59 in Chapter 3, curves *3* and *4* for steel and curves *1*, *2*, and *5* for non-ferrous metals are applicable to continuous furnaces as well as batch-type furnaces. In the heat treating of steel the rate of heat transmission to the stock must be low for metallurgical reasons. Compared to high-temperature furnaces, heat transmission is low as the result of the lower temperatures involved and the necessity for a low temperature gradient between the furnace and the stock. For the same temperature range, the rate of heat transmission to copper is higher than to bright steel, which, together with the lower specific heat and high conductivity of copper accounts for the fact that, in the range between 1300 F and 1600 F, copper can be heated at twice the rate of bright steel for equivalent results. In the process of heating copper for extrusion or other forming, a heating rate of 100 lb per (sq ft of hearth, hr) is an average value. The emissivities of both steel and copper vary with the condition of the surface.

5. LIQUID BATH FURNACES

The capacity of metal or salt baths for continuous operation differs from that of batch-type baths, because the movement of the metal through the bath increases the coefficient of heat transfer. J. D. Keller developed an empirical equation for galvanizing in a zinc bath:

$$k \text{ (Btu/sq ft, hr, F)} = 80 + 9V \text{ (velocity of stock in ft/min)}$$

The capacity of the bath also depends upon the purpose for which the heating bath is used. As shown in Fig. 75, the time required to heat wire in a metal bath is considerably less when the purpose is to coat the wire than it is for metallurgical purposes, where thorough heating of the wire is required.

6. CONTINUOUS FURNACES FOR TEMPERATURES BELOW 1400 F

As explained in Chapter 3, the extremely low intensity of radiation at temperatures below 1400 F causes difficulty in maintaining uniform temperatures throughout the furnace. This fact has led to the adoption of forced convection in the majority of the furnaces under discussion, as well as in furnaces of the batch-type. The heating capacity

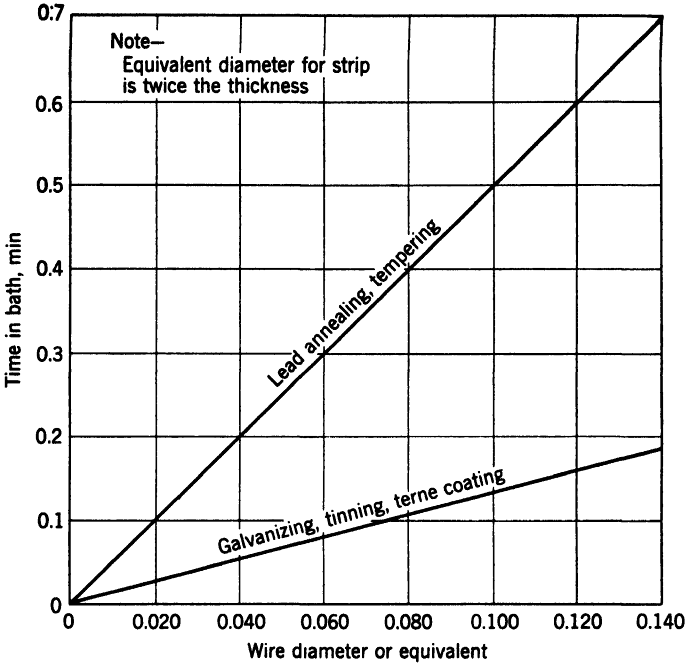


Fig. 75. Time required to heat steel wire or strip in bath of molten metal or salt. When heated for coating, the wire or strip is not thoroughly heated.

may, therefore, be calculated in the same manner, if consideration is given to the following differences.

If the conveying mechanism, including trays or other supports, is to be heated, its weight must be included in the weight of the charge. To prevent inflow of cold air at the ends of the furnace, a slight furnace pressure must be maintained. This requires not only a greater fan capacity but also a greater supply of fuel to the combustion chamber.

If the charge is stacked or hung in bunches, the phrase "pounds per square foot of hearth per hour" loses its significance, and Btu transmitted per square foot of charge per hour must be substituted. In many furnaces, within the temperature range under discussion, heat is derived from radiant tubes. By means of forced convection, heat is wiped off the tubes at a greater rate, so that the rating of tubes may be maintained the same as in furnaces at higher temperatures without affecting uniformity of temperature in the furnace. Many arrangements of radiant tubes are used in relation to recirculating fans, and, because of the great variety of designs, each case must be considered and calculated by itself.

Continuous furnaces for temperatures below 1200 F are always equipped with fans for recirculation. This means a practically constant furnace temperature. For that reason, heating capacity equals that of batch-type furnaces.

7. EXAMPLES

Example 1. Steel blooms 8 in. by 8 in. are to be heated in a pusher-type furnace to 2200 F—average temperature of blooms. Top heat only. The temperature difference between top and bottom of bloom is not to exceed 100 F. What is the heating time, and the heating rate in lb per (sq ft, hr)?

From Fig. 68, the required heating time is about 370 min. From the practical rule that heat travels ⅛ in. in 5 min, the heating time is 320 min. Since 8 in. of thickness equals about 320 lb per sq ft of steel-covered surface, a heating time of 320 min is equivalent to a heating rate of 60 lb per (sq ft, hr). A heating time of 370 min approximates a heating rate of 52 lb per (sq ft, hr). If the blooms cover 85 per cent of the hearth area, the heating rates are

$$0.85 \times 60 = 51 \quad \text{and} \quad 0.85 \times 52 = 44 \text{ lb per (sq ft of hearth, hr)}$$

In this example, the value of R does not matter because the rate of heating is determined solely by conduction in the stock.

Example 2. A single strip of skelp 15¾ in. wide by 0.39 in. thick passes continuously through an axial furnace of the type illustrated in Fig. 74. The total effective length of the furnace is 154 ft, and the total refractory area is 1540 sq ft. The skelp is to be heated from 100 to 2250 F with an average temperature of the furnace of 2450 F. Determine the capacity of the furnace.

$$\text{Ratio } R = \frac{2 \text{ sides} \times 1.31 \text{ ft} \times 154 \text{ ft}}{1540 \text{ sq ft}} = 0.26$$

By reference to Fig. 50 it can be seen that whenever the furnace temperature greatly exceeds the final temperature of the stock, the curve of temperature rise vs. time is practically a straight line, so that the mean of the temperature rise of the stock is one half of the temperature rise. Thus, the mean steel temperature in this furnace is 100 + ½(2250 − 100) = 1175 F.

From Table 5, for a gas temperature of 2450 F, stock temperature of 1175 F, $R = 0.26$, and $s = 2$ ft, and for a stock emissivity of 0.9, $k = 44$ Btu per (sq ft, hr, F). The difference in temperature is 2450 − 1175 = 1275 F, and the heat transfer is 44 Btu per (sq ft, hr, F) × 1275 F = 56,000 Btu per (sq ft, hr). This value is similar to that given in Fig. 71, curve *3*. The heat required is 380 − 12 = 368 Btu per lb.

The heating time is then

$$\frac{15.7 \text{ lb per sq ft} \times 368 \text{ Btu per lb}}{2 \text{ sides} \times 56{,}000 \text{ Btu per (sq ft, hr)}} = 0.052 \text{ hr}$$

or 3.1 minutes, and the production is

$$\frac{20.6 \text{ lb per ft} \times 154 \text{ ft}}{0.052 \text{ hr}} = 61{,}000 \text{ lb per hr}$$

As stated in the text, the center part of the strip is colder than 2200 F, whereas the edges reach a higher temperature than 2450 F.

EXAMPLE 3. A walking beam furnace 4.3 ft wide by 16 ft long is to heat automobile axles from 100 to 1600 F. There will be 60 axles in the furnace at any one time; each axle weighs 51 lb and has 1.9 sq ft of surface exposed to the heat. Refractory area of the furnace is 280 sq ft. Furnace temperature is 1650 F, and the furnace is fired from both sides above the axles, with circulation of the gases below the axles. Calculate the production and rate of heating.

$$\text{Ratio } R \text{ is } \frac{60 \text{ axles} \times 1.9 \text{ sq ft}}{280 \text{ sq ft}} = 0.41$$

From Fig. 52 for stock temperature of 1600 F, $R = 0.41$, and emissivity of the stock = 0.6, $k = 0.65 \times 16 = 10.3$ Btu per (sq ft, hr, F). The difference in temperature between the furnace and stock is $\frac{1}{3}(1600 - 100) + 50 = 550$ F.

The rate of heat transfer is 10.3 Btu per (sq ft, hr, F) × 550 F = 5680 Btu per (sq ft, hr), which compares closely with the equivalent value in Fig. 59, curve 3, for heat treating where $R = 0.41$. The heat required per pound of steel is 265 − 12 = 253 Btu.

The heating time is

$$\frac{51 \text{ lb} \times 253 \text{ Btu per lb}}{1.9 \text{ sq ft} \times 5680 \text{ Btu per (sq ft, hr)}} = 1.2 \text{ hr}$$

The production is

$$\frac{60 \text{ axles} \times 51 \text{ lb}}{1.2 \text{ hr}} = 2550 \text{ lb per hr}$$

The heating rate is

$$\frac{2550 \text{ lb per hr}}{4.3 \text{ ft} \times 16 \text{ ft}} = 37.0 \text{ lb per (sq ft of hearth, hr)}$$

In this example, the total area of each axle is effective, because the alloy walking beams place the axles on alloy piers at each step, with ample circulation and radiation to all parts of the axle.

chapter **5**

The Fuel Economy of Furnaces

1. GENERAL STATEMENT ON FURNACE EFFICIENCY

The words "economy" and "efficiency," when used in their true sense in connection with industrial furnaces, have reference to the heating cost per unit weight of finished, saleable product. "Heating cost" includes not only the cost of fuel but also the cost of firing and superintending the furnace, amortization of furnace, the costs of maintenance and repair, the cost of generating a protective atmosphere, and the cost of burned, spoiled, or otherwise rejected pieces. Furthermore, it includes the cost of machining those pieces which, at inspection, are found to be defective because of improper heating. Finally, it includes the cost of handling the material into and out of the furnace.

With so many different items entering into the cost of heating, it is quite possible that, in some cases, the highest-priced fuel or other source of heat energy may be the cheapest in the end, so far as the total heating cost is concerned. As a matter of fact, this very condition exists in many electrically heated furnaces for heat treating. This phase of the problem is dealt with in Volume II (section on critical comparison of fuels and of furnace types).

In the heating of some metals, for instance steel, part of the heat is furnished by combustion (oxidation) of the charge. The heat of combustion of iron or steel is 2420 to 3240 Btu per lb of iron, depending on the kind of oxide formed, with an average of 2850 Btu per lb. If steel is heated to 2200 F, from 1 to 4 per cent of the weight of the charge is usually lost by scaling (which constitutes the "furnace loss"). In consequence, 57,000 to 230,000 Btu per net ton of steel have their origin in the combustion of part of the charge. It will be shown later that this is a minor fraction of the heat (or fuel) consumption of a furnace; however, this fraction is very expensive, because steel, with 2850 Btu per lb, costs more than ten times as much per ton as coal, with 12,000 to 14,000 Btu per lb.

In spite of the statements made in the first paragraph of this chapter, we encounter in practice many furnaces in which fuel cost is the outstanding item of expense and in which fuel economy is worthy of study, wholly aside from the other factors making up the sum total of heating cost. Moreover, it is necessary to know beforehand the probable fuel consumption of a furnace, in order to select the correct size and number of burners; to dimension ports, vents, and stack; and to select auxiliary equipment of correct size.

In the study of the fuel economy of industrial furnaces, the student is usually astonished by their low thermal efficiency. Whereas boiler efficiencies range from 60 to 90 per cent, or even slightly higher, furnace efficiencies are sometimes as low as 5 per cent in combustion types, and not much, if any, higher in electric types, if the thermal efficiency of generation of electric energy is included in the figure.

With favorable conditions, with excellent (and usually expensive) design, and with good operation, furnace efficiencies up to 60 per cent, or even slightly higher, can be obtained, but such high values are the exception rather than the rule. By "efficiency" is here meant "fuel efficiency," which is the ratio of the heat input into the stock to the potential heat in the fuel, measured in the long run. Tabular data on fuel efficiency and fuel consumption are given at the end of this chapter.

One reason for the difference in thermal efficiency between boiler furnaces and industrial furnaces is the final temperature of the material being heated. Gases can give up heat to the charge only as long as they are hotter than the charge. In consequence, the flue gases must leave industrial furnaces at a very high temperature except for a short time after a cold start. On the basis that the products of combustion leave an industrial furnace at a temperature of 50 F in excess of that of the charge, two (almost straight) curves have been shown in Fig. 76; they indicate the highest possible thermal efficiency of industrial furnaces as a function of furnace temperature. In fuelfired furnaces, the thermal efficiency cannot be improved, unless expensive equipment is installed for utilizing the heat of the outgoing products of combustion in some manner. Experience has shown that furnaces with expensive heat-salvage equipment must be intelligently and carefully operated if the desired high thermal efficiency is to be maintained; and such a method of operation necessitates the employment of well-paid attendants.

The curves for maximum possible efficiency, as shown in Fig. 76, were calculated for simple furnaces with no preheat of charge, fuel, or air. As a rule, the thermal efficiency of industrial furnaces lies much below the values indicated by the ideal curves. The discrepancy is particularly noticeable in low-temperature furnaces. The reason for this discrepancy lies in the fact that the products of combustion must be cooled down before they are allowed to come in contact with the charge. Otherwise, the latter will be seriously overheated or burned in spots. In this respect the industrial furnace differs from the boiler furnace.

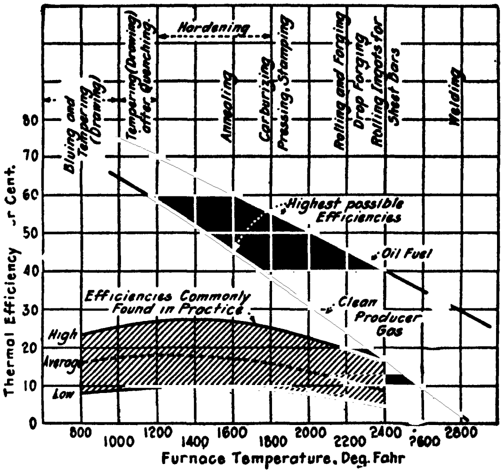


Fig. 76. Ideal and actual efficiencies of industrial furnaces of the intermittent or batch type.

In the boiler furnace, it is quite difficult to overheat the tubes. In industrial furnaces, cooling of the products of combustion is often accomplished either by excess air, or by contact with cold surfaces, or by both methods—either one of which is wasteful.

2. DISTRIBUTION OF HEAT IN A FURNACE

For the sake of a clear understanding of the distribution of heat in a simple furnace, Fig. 77 should be consulted. Most of the heat is released in the combustion zone at the left and travels from there to the right. The passage of heat into the stock, as indicated by arrows 1, is desired. But heat also goes elsewhere; some of it passes into the furnace walls and some into the hearth, as indicated by arrows 3, increasing the temperature of those parts. Another portion of the heat is lost to the surroundings by radiation and convection from the outer surface of the walls, or by conduction into the ground; see arrows 2. Through cracks or other openings, heat radiates away, 4; and furnace gases pass out around the door 5, frequently burning in the open and carrying off heat. Heat is lost every time a door is opened. Then there are special losses, peculiar to certain types of furnaces; for example, in furnaces that heat only part of a long piece of metal, heat passes out along the metal from the part which is in the furnace to the part which is outside, and is there dissipated to the surroundings 6. Radiation from the exposed liquid surface of salt baths or lead baths, and conduction of heat through the terminals or electrodes of electric furnaces, are other cases in point. Water cooling of skid pipes and of conveyor rollers absorbs large quantities of heat and lowers thermal efficiency. If the charge is heated in containers or on traveling chains a large part of the heat is dissipated in the open by these devices, after they have been brought up to furnace heat. Finally, heat passes out with the products of combustion 7, either in the form of sensible heat, or as undeveloped heat of combustibles escaping unburned (incomplete combustion).

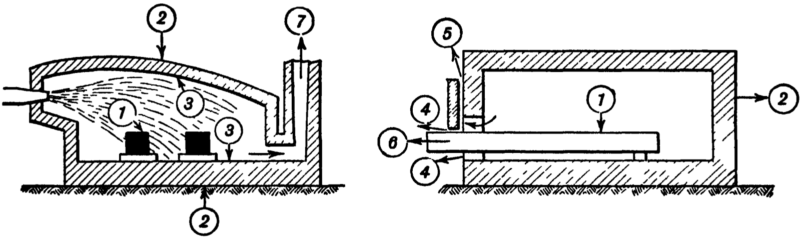


Fig. 77. Flow of heat in a furnace.

Fuel economy demands that the fraction of the total heat that passes into the stock be as large as is compatible with correct heating. The first step toward attainment of this goal requires the solution of two problems: first, determination of the heat losses and the methods of reducing them; second, determination of the quantity of fuel or of electric energy that is required for heating a given amount of metal to a given temperature in a given furnace. In conformity with this plan, the heat losses are now taken up in order.

3. HEAT LOST FROM FURNACE WALLS (WALL LOSS)

The heat lost from the outside of furnace walls is one of the salient features affecting the economy of furnaces and must be discussed at length. The wall loss during continued, uninterrupted operation of a furnace differs from the wall loss of the same furnace if the latter is operated intermittently. The wall loss during continuous operation is discussed first.

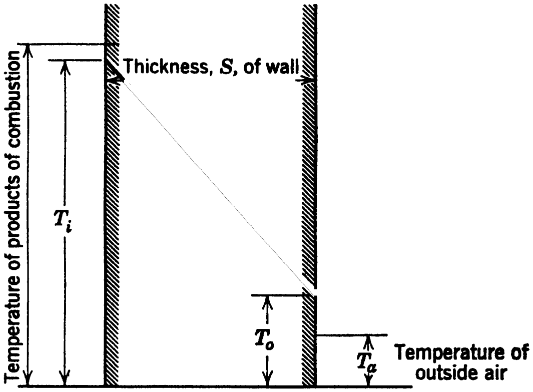

Temperature of products of combustion
Thickness, S, of wall
T_i
T_o
T_a
Temperature of outside air

Fig. 78. Temperature gradient for steady flow of heat through a furnace wall, with equal pressure on both sides of the wall.

(a) Wall Loss during Steady Operation. For steady flow of heat through a wall, the conditions indicated in Fig. 78 are used. The temperature of the products of combustion exceeds the temperature of the interior surface of the wall. The temperature of the products of combustion is very seldom known, because of the difficulties encountered in measuring the temperature of gases. The temperature of the inner face of the wall can be measured more easily and is, as a rule, known. The temperature of the wall drops steadily toward its outer surface, at which place the temperature exceeds that of the surrounding air. The heat loss for a given extent of wall and for a given furnace temperature becomes less if the wall is made thicker, or if the wall is made of a better insulator, or if the outer wall surface is of such a character that it does not readily give up its heat to the surrounding media. Mathematically these relations are expressed by the equation

$$Q = \frac{C(T_i - T_o)}{s} = k(T_o - T_a) \qquad (13)$$

where Q = the heat transmitted in Btu per (sq ft, hr)
T_o, T_a, T_i = temperatures (see Fig. 78)
C = conductivity of material of wall, Btu per (ft, hr, F)
k = coefficient of heat dissipation from outer surface of wall, Btu per (sq ft, hr, F)
s = thickness of wall, feet

The unknown quantities in this double equation are Q and T_o. They can be found if all the other quantities are known. Unfortunately for simplicity of calculation, neither k nor C is constant; they vary according to several circumstances.

The coefficient k varies with the temperature of the outside wall, with its location (whether horizontal or vertical), also with its physical condition, and probably with its size. This coefficient is not a true coefficient of heat transfer; it is a pseudocoefficient, for this reason: The heat is dissipated from the walls both by radiation and by convection. No true coefficient can correctly take care of a composite function.

Although different materials have different radiating power (emissivity), it is fortunate that there is very little difference in the coefficient of dissipation for the different kinds of surface commonly found on the outside of furnace walls. For that reason, the values of Fig. 79 are acceptable for calculating purposes. The coefficients of Fig. 79 are sufficiently accurate for brick surfaces and for steel surfaces.

The emissivity of walls that were coated with aluminum paint is lower than the emissivity of bricks. For newly painted surfaces $e = 0.20$. An average value for design purposes is $e = 0.28$. The coefficient for radiation plus convection from aluminum-painted walls was also drawn in Fig. 79.

For calculation purposes, it is sometimes convenient to express the heat loss as total quantity of heat transmitted per unit area, in a given time, for different surface temperatures and for an average temperature of surroundings. Figure 80 gives such values for rough brick surfaces and for walls that are coated with aluminum paint, and for 100 F temperature of surroundings. The values were calculated from the equation

$$Q = k\left[\left(\frac{T_o + 460}{100}\right)^4 - (5.6)^4\right] + 0.28\,(T_o - 100)^{5/4}$$
$$= \text{Btu per (sq ft, hr)} \qquad (14)$$

For rough vertical surfaces, $k = 0.155$. For medium-old aluminum

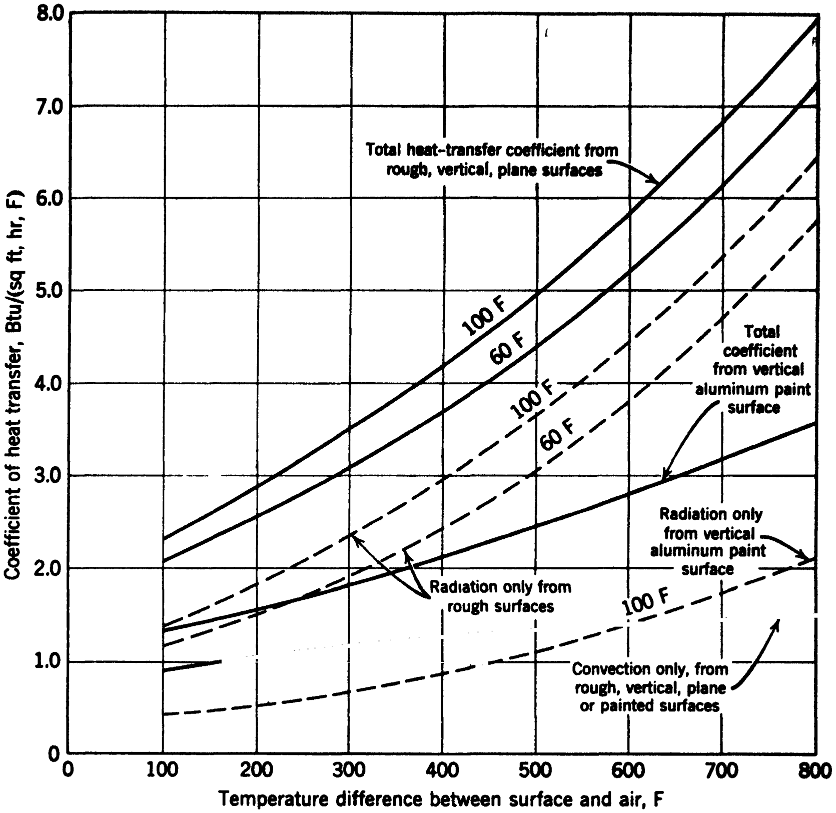


Fig. 79. Coefficients of heat dissipation for different conditions in still air. Numbers on curves denote temperatures of surrounding air.

paint, $k = 0.05$. Air circulation (draft) increases the convection coefficient 0.28.

Figure 80 may give the erroneous impression that much heat is conserved by coating the cold face with aluminum paint. Actually, the saving of heat is much smaller than is indicated by the illustration. Heat does not readily leave shiny surfaces. In consequence T_0 is higher for shiny surfaces than it is for rough surfaces. For an example, see below.

Coefficients C of conductivity of refractories likewise increase with temperature, but not nearly as rapidly as the coefficient of heat dissipation. From available test data, the curves of Fig. 81 were selected as average values. Insulating brick selected for the calculation is of the 2000 F type. (See Fig. 358 for other insulations.)

With the values of k contained in Fig. 79 and those for C in Fig. 81, equation 13 can be solved for heat loss Q and outside wall temperature T_o, the only difficulty being that each case can be solved only by a series of approximations. Values must be assumed for C and k, so that T_o may be computed. If the result shows that the value used for k is different from the one belonging to T_o, or that the value of C which has been used is different from the one belonging to the average between T_i and T_o, then new values must be selected for C and k and the calculation must be repeated. Although this guessing at the proper values of C and k is very troublesome in making a single calculation, it becomes less difficult after a large number of calculations have been made. In selecting the conductivity C, an average value is used be-

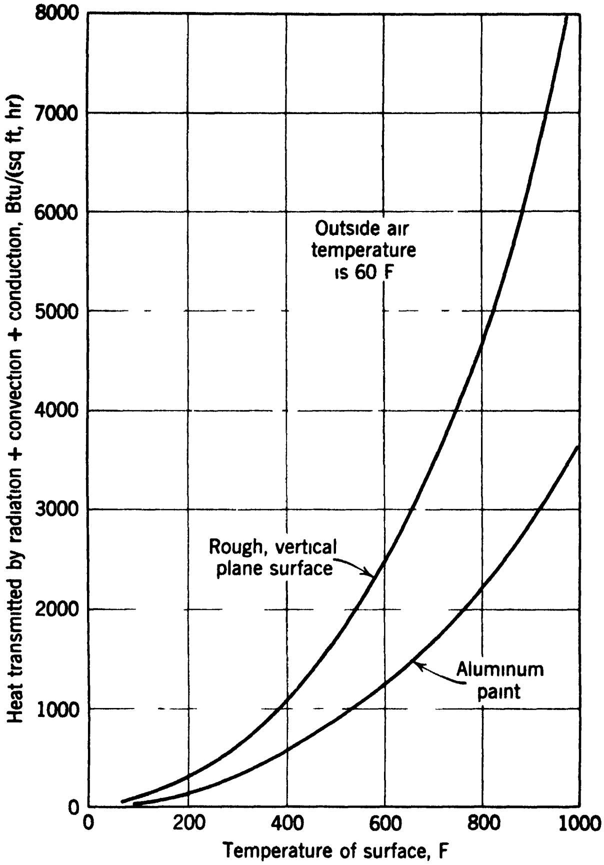


Fig. 80. Heat losses from rough vertical plane surfaces, with and without aluminum paint.

tween the conductivities at the hot side and those at the cold side of the wall.

By the method indicated in the preceding paragraphs, it is possible to compute and plot curves showing heat losses, as well as outside surface temperatures, for given hot-face temperatures and wall thicknesses. Before the calculation of such curves is discussed, the method of computing Q and T_o for a composite wall should be explained. The space-temperature diagram for such a wall assumes the shape indicated in Fig. 82, and the equation of heat flow takes the form:

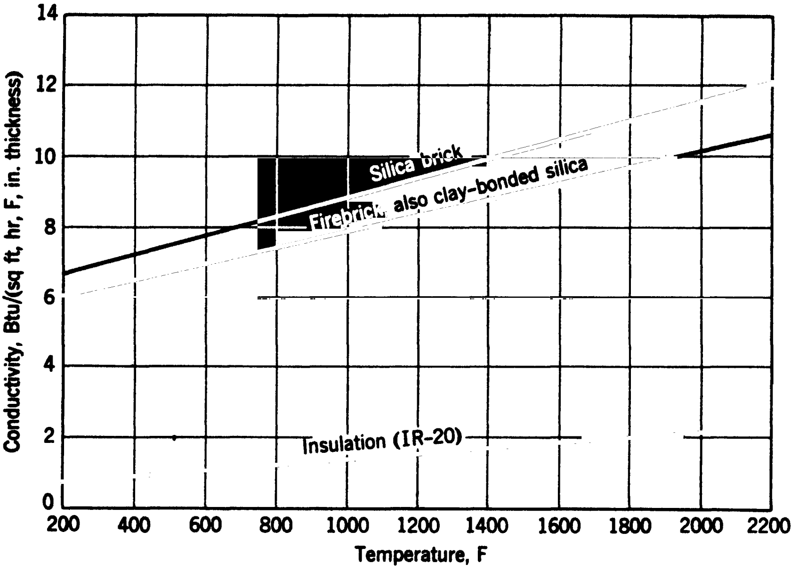


Fig. 81. Average conductivity of refractory materials (see also Figs. 357 and 358).

$$Q = \frac{C_1}{s_1}(T_i - T_b) = \frac{C_2}{s_2}(T_b - T_o) = k(T_o - T_a) \tag{15}$$

This triple equation is solved in the following manner:

$$\frac{Qs_1}{C_1} = T_i - T_b$$

$$\frac{Qs_2}{C_2} = T_b - T_o$$

$$\frac{Q}{K} = T_o - T_a$$

By addition:

$$Q\left(\frac{s_1}{C_1}+\frac{s_2}{C_2}+\frac{1}{k}\right)=T_i-T_a$$

$$Q=\frac{T_i-T_a}{\frac{s_1}{C_1}+\frac{s_2}{C_2}+\frac{1}{k}} \tag{16}$$

An example will illustrate the use of this equation.

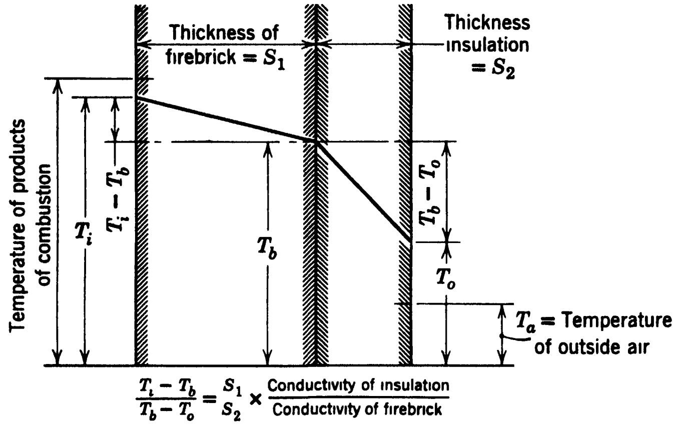

Thickness of firebrick = S_1
Thickness insulation = S_2
Temperature of products of combustion
T_i
$T_i - T_b$
T_b
$T_b - T_o$
T_o
T_a = Temperature of outside air
$\frac{T_i - T_b}{T_b - T_o} = \frac{S_1}{S_2} \times \frac{\text{Conductivity of insulation}}{\text{Conductivity of firebrick}}$

Fig. 82. Temperature gradient for steady flow of heat through a composite wall, with equal pressure on both sides of the wall.

Example. Let it be required to compute the amount of heat transferred in one hour through a furnace wall made up of 9 in. of firebrick and 2½ in. of IR-20 (2000 F) insulating refractory bricks. The temperature of the inner side of the wall = 1800 F; temperature of outer air = 60 F.

As shown in Fig. 81, the conductivity of firebrick and, in fact, of almost all refractory materials grows with temperature. A first estimate of the probable temperature T_b must, therefore, be made, and the actual heat transfer must be computed with the values of heat conductivity applying to the estimated temperature range.

For the first trial take temperature $T_b = 1310$ F. From Fig. 81, average conductivity of firebrick for the range from 1310 to 1800 F (average = 1555 F) is 9.0 Btu per (sq ft, hr, F) and 1 in. of thickness. Take $T_o = 250$ F; then average conductivity of IR-20 insulating bricks for the range 250 to 1310 F is 1.10. Substitution of numerical values and of 2.6 from Fig. 79 in equation 16 furnishes

$$\frac{1800-60}{\frac{9}{9.0}+\frac{2.5}{1.1}+\frac{1}{2.6}} \qquad \frac{1740}{1.0+2.28+0.38}=476 \text{ Btu per (sq ft, hr)}$$

For a check on the correctness of the assumed temperature range, T_o and T_b must be calculated from equation 15. From $Q = k(T_o - T_a)$ follows

$$T_o = T_a + \frac{Q}{k} = \frac{476}{2.6} + 60 = 243 \text{ F}$$

Similarly, $T_b = T_i - \dfrac{Qs_1}{C_1} = 1324$ F.

The calculated temperatures differ slightly from the assumed temperatures. The difference is, however, so small that a recalculation would furnish practically the same value as the first calculation. In view of the uncertainty of the values for thermal conductivity, a recalculation is not necessary.

Effects of Aluminum Paint. The foregoing example was calculated for a rough outside wall surface, with the value of k taken from Fig. 79. If the surface is coated with aluminum paint, a different value of k will apply, and since the aluminum paint (by reducing radiation) is an insulator, the temperatures of both the outside surface and the interface between firebrick and insulation can be expected to be greater than with the rough surface. After several trial calculations we judge that the outside temperature is 330 F and that the interface temperature is 1320 F.

The average conductivity of firebrick for an average temperature of 1560 F is 9.1 Btu per (sq ft, hr, F) and 1 in. of thickness; and of the insulation for an average temperature of 825 F, it is 1.18. From equation 14 for 60 F air temperature and with a factor of 0.05 for aluminum paint

$$k = \frac{0.05\left[\left(\frac{330 + 460}{100}\right)^4 - \left(\frac{60 + 460}{100}\right)^4\right] + 0.28(330 - 60)^{5/4}}{330 - 60}$$

$$= 1.72 \text{ Btu per (sq ft, hr, F)}$$

By substitution of these values in equation 16, $Q = 472$ Btu per (sq ft, hr). To check the assumed temperatures,

$$T_o = \frac{472}{1.72} + 60 = 335 \text{ F} \quad \text{and} \quad T_b = 1800 - 472 \times 0.99 = 1332 \text{ F}$$

The effect of aluminum paint on heat passing through the wall is surprisingly small. The shiny surface of the cold face, by radiating less heat, for a given temperature, acts as an insulator and raises the temperatures of the firebrick and of the insulating refractory. The thermal conductivity of both refractories rises with temperature, with the result that almost as much heat escapes through the shiny wall as through the dull wall. However, the paint reduces gas-leakage through the wall.

The method of first assuming temperatures, then calculating heat flow, and thereafter checking for correctness of assumed temperatures is a tedious process. There is no reason why this calculation should be repeated time and time again. It is preferable to calculate the wall losses once for all and to lay them down in tabular or graphical form. A method of calculating wall losses is explained in the following paragraphs.

For each inside surface temperature and each wall thickness, a curve is plotted, showing heat conducted through the wall for various outside temperatures of wall surface. The abscissae are quantities of heat transmitted through unit area in unit time: the ordinates are temperatures. In Fig. 83, such curves are shown for an inside temperature of 2600 F, and four different thicknesses of firebrick wall. If, as another example, a set of curves is to be drawn for 1800 F inside wall temperature, then the heat flow for a 9-in. wall with 600 F temperature of outer surface is calculated as follows: Average temperature = $\frac{1}{2}(1800 + 600) = 1200$ F; average conductivity = 8.3; and rate of heat flow $= \frac{8.3}{9}(1800 - 600) = 1108$ Btu per (sq ft, hr). Although a different conductivity must be used for each point along a curve, the curves are comparatively simple to calculate, because the abscissae are inversely proportional to the wall thickness. Thus, the abscissae for the 4½-in. wall are twice those for the 9-in. wall. On the same curve sheet is plotted the curve showing heat emitted per square foot of outside surface and per hour, as a function of outside temperature of wall (see also Fig. 80). It is evident that the intersections of this curve with the curves of heat flow through the brick furnish values of heat loss

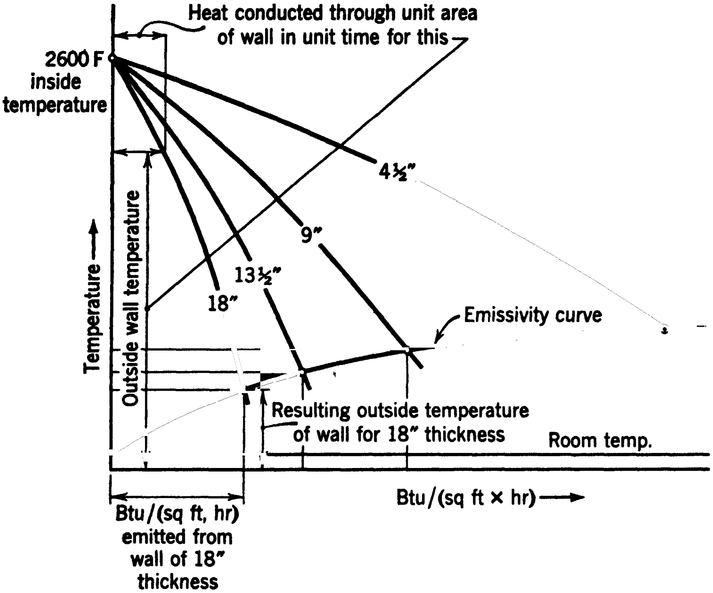

Heat conducted through unit area of wall in unit time for this
2600 F inside temperature
4½"
9"
13½"
18"
Temperature
Outside wall temperature
Emissivity curve
Resulting outside temperature of wall for 18" thickness
Room temp.
Btu/(sq ft × hr)
Btu/(sq ft, hr) emitted from wall of 18" thickness

Fig. 83. Graphical method of calculating heat losses through homogeneous walls.

and of outside wall temperature. It may be repeated that a different set of curves must be calculated for each inside temperature of wall. However, four or five sets of curves furnish enough points to cover the whole field of industrial furnaces.

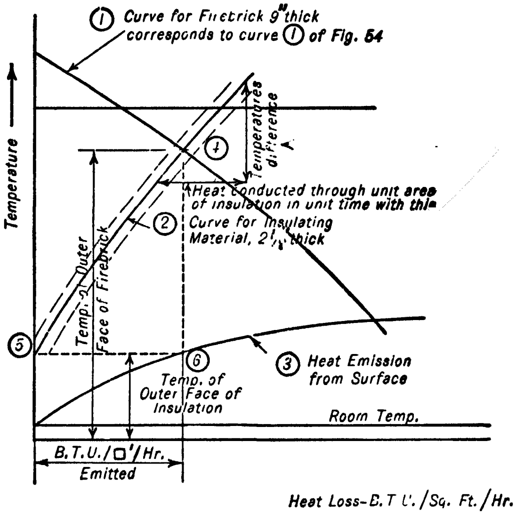

① Curve for Firebrick 9" thick corresponds to curve ① of Fig. 54
Temperature
Temperatures difference
④
Heat conducted through unit area of insulation in unit time with this
② Curve for Insulating Material, 2½" thick
Temp. of Outer Face of Firebrick
⑤
⑥
Temp. of Outer Face of Insulation
③ Heat Emission from Surface
Room Temp.
B.T.U./□'/Hr. Emitted
Heat Loss-B.T.U./Sq. Ft./Hr.

Fig. 84. Graphical method of calculating heat losses through composite walls.

The graphical method of Fig. 84, for composite walls, requires the use of transparent paper or tracing cloth for plotting the curves. Curve 1 (for the firebrick) and curve 3 (heat emission from surface) are both plotted on the same curve sheet. Curve 2 (for the insulating material) is plotted, in a reverse direction, on a separate, transparent sheet which can be moved on the other sheet. Curve 2 is moved to right or left, but horizontal temperature lines on upper curve sheet must always coincide with the corresponding ones on the lower sheet. Curve 2 is adjusted so that the vertical line from 4 and the horizontal from 5 will intersect at 6 on curve 3. Abscissa 5–6 represents the heat loss, and points 4 and 6 represent temperature of outer brick surface and of outer surface of insulating material, respectively.

By these methods, the curve sheets, Figs. 85 and 86, were computed and plotted. The former shows temperatures of the outside wall surface as a function of furnace temperature and wall thickness, whereas the latter gives the heat loss per square foot and hour as a function of the same independent variable.

Figures 85 and 86 were plotted for dense firebrick. Some walls of industrial furnaces are built of dense firebrick which is used entirely

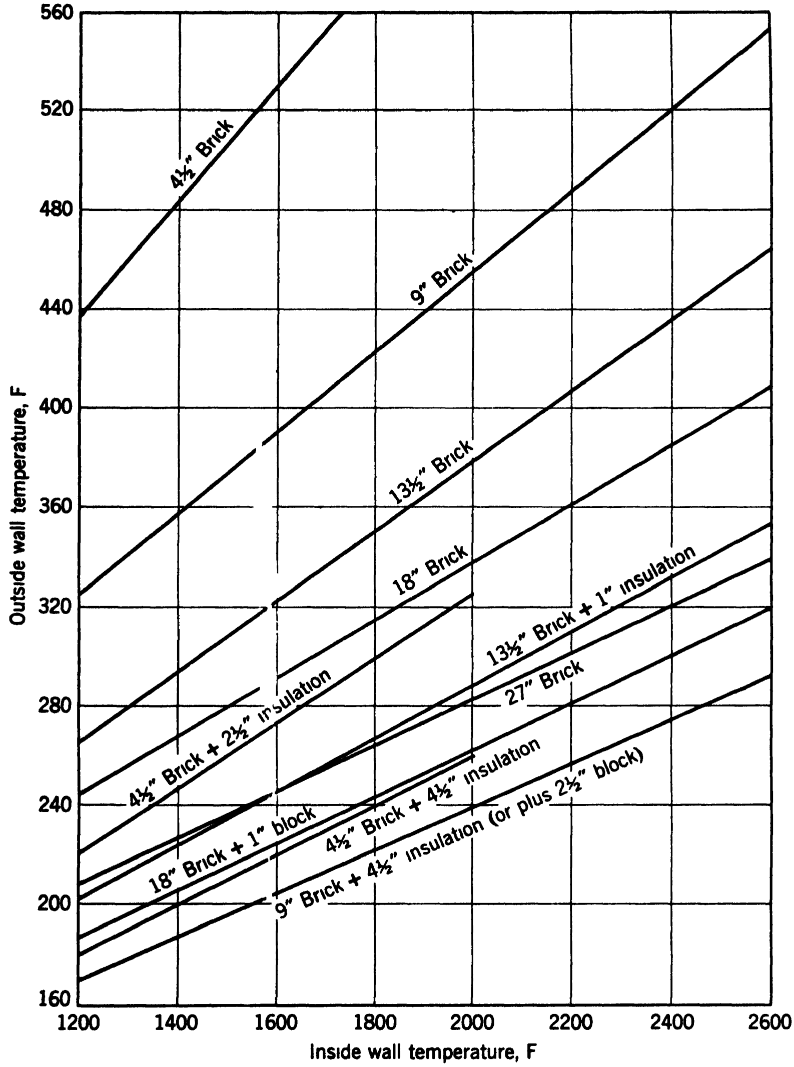


Fig. 85. Temperature relation of faces of furnace walls (for atmospheric pressure in furnace). Insulation is 2000 F insulating firebrick (IR-20); block is calcined diatomaceous silica (Super X or equivalent).

for furnaces of several other types. Many furnace walls are now made of rammed fireclay. The heat loss of walls made of rammed fireclay is lower than the heat loss shown in Fig. 86, because the thermal conductivity of plastic clay is, after drying and burning in place, about 60 per cent of the conductivity of dense firebrick. The lower con-

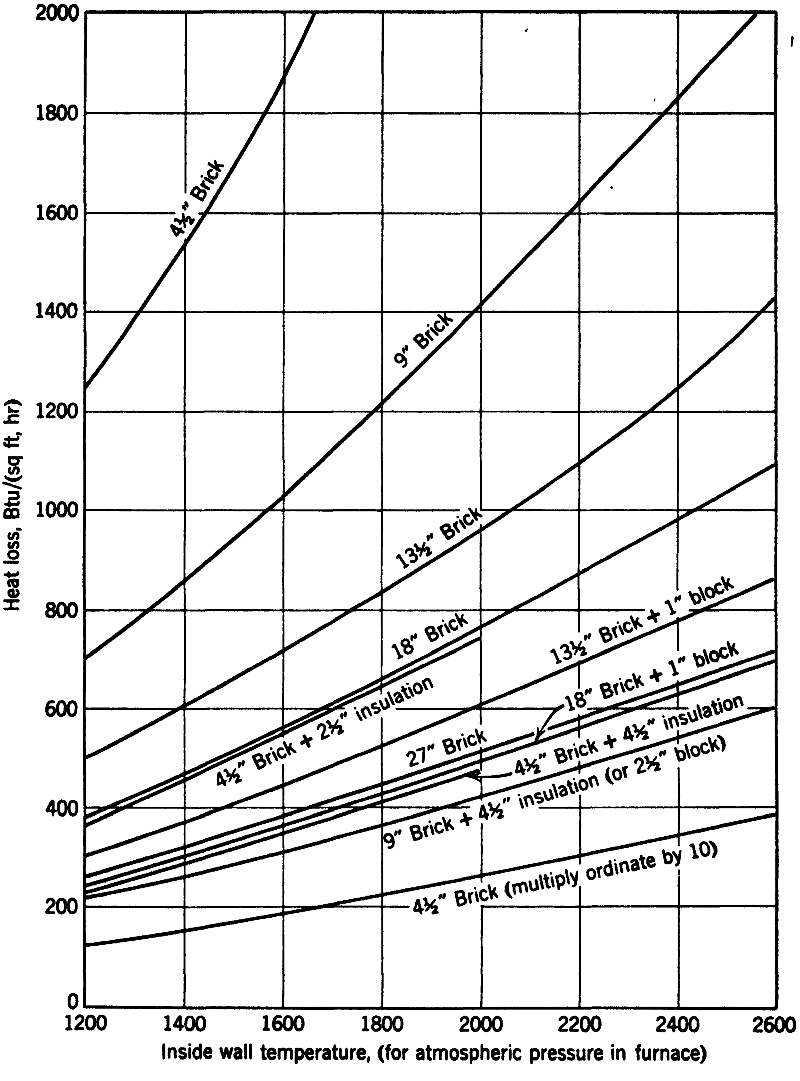


Fig. 86. Heat losses through furnace walls (for atmospheric pressure in furnace). In practice, losses may be increased as much as 30% over these values. Insulation is 2000 F insulating firebrick (IR-20); block is calcined diatomaceous silica (Super X or equivalent).

ductivity does not bring about a corresponding reduction of wall loss, because the plastic is rammed into spaces between anchors of dense firebrick. Roughly, firebrick walls of thickness "one" and rammed walls of thickness 0.7 have equal heat loss.

The majority of industrial furnaces are constructed of insulating refractories. For that reason, Fig. 87 was computed.

In order to make the illustration easily read, only three types of insulating refractories (IR-20, IR-23, and IR-28) were used in plotting the curves. For data on thermal conductivity of other insulating refractories Fig. 358 may be consulted.

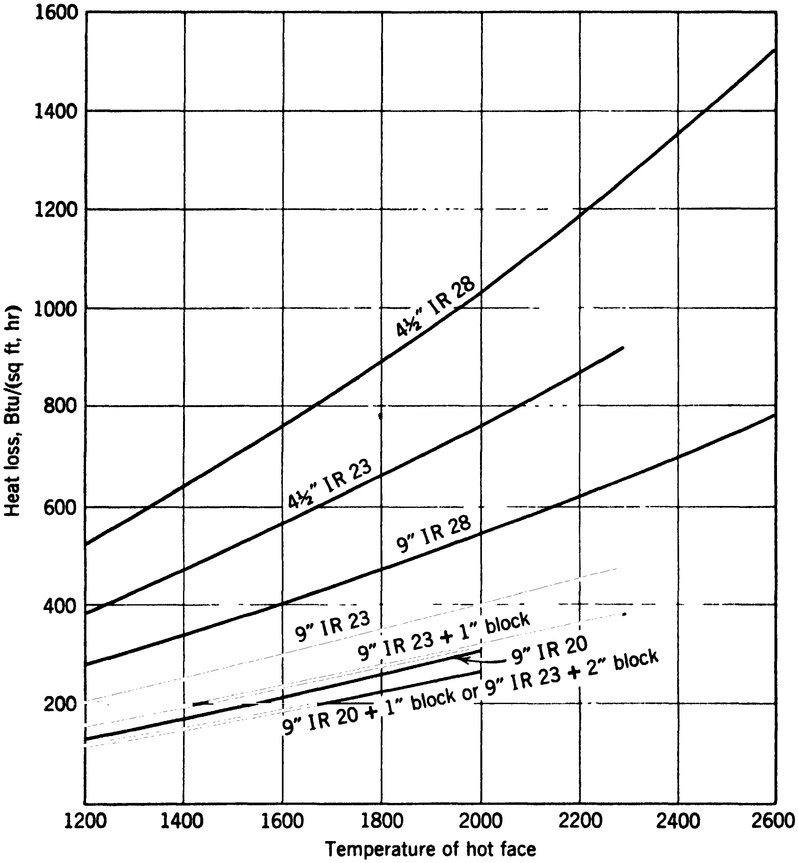


Fig. 87. Heat losses through furnace walls of lightweight firebrick.

IR = Insulating refractory brick.

Numbers 20, 23, and 28 indicate maximum recommended temperature, $\frac{\text{deg F}}{100}$

Block = calcined diatomaceous silica (Super X or equivalent).

It must be understood that the charts for wall loss are based on equal pressures on both sides of the walls. If the pressures differ, gaseous fluid flows, leaks, or seeps toward the region of lower pressure. Notable pressure difference exists in roofs and sidewalls of tall furnaces, in regenerators, uptakes, and stacks. For a given pressure difference and a given thickness of wall, the flow depends mainly on the width and the number of the joints between the bricks.

Tests on permeability (gas-passability) were made on silica bricks in Germany.* Bansen found that for equal conditions of brick thickness and pressure difference, leakage was in some cases 20 times greater than in others. Veit established the equation

$$\text{Leakage (cu ft/hr)} = f \times \frac{\text{Area of ref. (sq ft)} \times \text{Pressure diff. (in. water)}}{\text{Brick thickness (in.)}}$$

The leakage is referred to 60 F. For bricks and tiles without joints, f was found to be 70. Coating the bricks (probably with sodium silicate) reduced the permeability. For brickwork consisting of 90 per cent brick area and 10 per cent mortar-filled joints, f rose to 3600.

The introduction of insulating refractories brought up the problem: How does their permeability compare to the permeability of dense bricks? The answer was furnished by tests made at the National Bureau of Standards.† For various brands of dense firebrick, the factor f was found to lie between 2.4 and 19; whereas, for insulating refractories, f was found to lie between 200 and 550. The joints between adjacent bricks introduce an unknown multiplier.

So far as fuel economy is concerned, seepage of gas through walls is of little importance, if the brickwork has been laid with care and if the bricks are flat and square. An example will illustrate this statement.

In the heating of steel to 2200 F, a rate of heating 60 lb of steel per square foot of hearth times hour is common, and so is a heat consumption of 3,000,000/2000 = 1500 Btu per lb of heated steel. With natural gas, the combustion of 1.5 cu ft releases that quantity of heat. Since 1 cu ft of natural gas produces 10 cu ft of products of combustion, the flow of gases (cold measure) per square foot of hearth equals 60 × 15 = 900 cu ft per hr. Let the leaking wall area (sidewalls and roof) equal three times the hearth area and let the average furnace pressure equal 0.025 in. of water. With a rather high factor of $f = 2700$, the leakage through a 13½-in. wall equals $\frac{2700 \times 3 \times 0.025}{13\frac{1}{2}} = 15$ cu ft per hr, which is less than 2 per cent of the total gas flow.

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* H. Bansen, "Gasdurchlässigkeit van Silikasteinen, Mörtelfugen und Mauerwerk in Siemens-Martinöfen," *Archiv für das Eisenhüttenwesen,* 1927–1928, pages 687 to 692.

P. Veit, "Untersuchungen über den Weg der Becheizungsgase in Koksöfen, *Stahl and Eisen,* 1953, pages 798 to 799.

† Massengale, Mong, and Hendle, *Journal of the American Ceramic Society,* 1953, page 274.

Gas leakage increases the wall loss, but it would be wrong to add the wall loss by leakage to the wall loss by conduction, expressed in Figs. 86 and 87, because leakage increases the temperature of the cold face of the wall and thus reduces heat flow by conduction.

Although the permeability of well-built walls does not seriously affect heat economy, several facts should be mentioned. Figures 86 and 87 were computed for average bricks in gastight walls. But not only does the thermal conductivity of bricks vary, but gas leakage exists. Repeated heating and cooling of a furnace loosens the bond between bricks, causes cracks and produces spalling. In consequence, furnace engineers increase the values of Figs. 86 and 87 as much as 20 per cent.

Heat loss by conduction and leakage is greatly reduced by enclosing the furnace in a steel jacket. This practice is common on vertical walls and on the roofs of furnaces which carry protective atmospheres. The roofs of high-temperature furnaces are covered with about 1 in. of fibrous cement or by a glassy foam. Both materials are claimed to be impervious.

The charts for flow of heat through furnace walls were calculated for walls of infinite extent, with area of hot face equal to area of cold face. A simple way of taking care of the difference in areas is to figure with an area midway between hot face and cold face. In view of the uncertainties of physical values, such as thermal conductivity, emissivity, and permeability, the method of using an intermediate area is sufficiently accurate for the majority of practical cases. In very small furnaces, the ratio of outer to inner area is so great that "edges' effect" is noticeable. The effect of edges is illustrated by Fig. 88, which is due to Langmuir.* The illustration shows isothermals and lines of heat flow. The isothermals near the inner wall surface lie closer together as the corner is approached. This means that the temperature gradient is greater near and at the corner and that the rate of heat flow is also greater. This additional heat loss can be accounted for by taking the equivalent heat-transmitting area to be larger than the average between cold face and hot face. This situation is encountered in small, thick-walled furnaces. With small furnaces, experimentation is less expensive than calculations.

* *Transactions American Electrochemical Society,* 1913, page 53.

Figures 86 and 87 may be used for roofs, with an accuracy that is sufficient for practical purposes. Figure 86 may be used for heat flow through ventilated hearths, if the latter consist of firebrick.

Non-ventilated hearths, that is to say, hearths resting on the ground or on a heavy foundation, are seldom used in large metal-heating furnaces that carry high temperatures (2000 F or higher). Solid hearths serve in small industrial furnaces and in large furnaces of other types. The heat loss through solid hearths is usually less than the loss from an equal area of sidewall, but is not as small as has sometimes been supposed. A complete discussion of the subject was given by J. D. Keller in *Transactions American Society of Mechanical Engineers,* Volume 50, 1928, page 11.

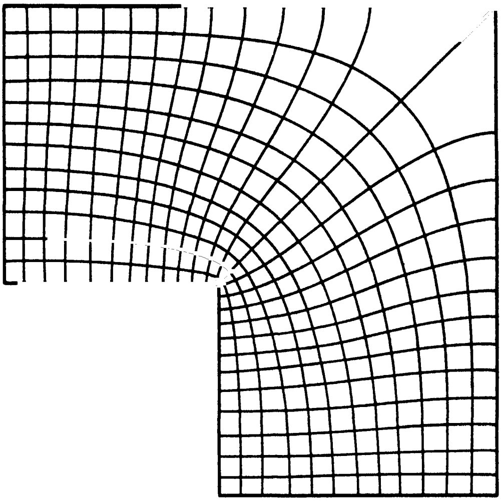

Fig. 88. Temperature distribution in a corner.

The isothermals and the lines of heat flow under the hearth are shown in Fig. 89. The numbers on the isothermals denote the fraction of the total temperature difference between the interior of the furnace and the surroundings; thus, for a 2200 F furnace and 60 F room temperature, the isothermal marked 0.9 would correspond to an actual temperature of $60 + 0.9(2200 - 60) = 1986$ F. The temperature penetrates deepest under the center of the hearth. For furnaces operated steadily over long periods of time, therefore, in order to protect the concrete foundation from being overheated, the brick hearth must be made quite thick (30 to 50 per cent of the hearth width).

The total heat loss is proportional not to the area but to the linear dimension of the hearth (for hearths of the same geometrical shape, in which length and depth may be considered as functions of width). The loss, after practically steady temperature conditions have been reached, is

$$Q = S \cdot C \cdot A \cdot (T_i - T_a)/D \qquad (17)$$

where Q is the heat flow in Btu per hour through a hearth of area A sq ft maintained at temperature T_i, when the temperature of the outside air is T_a; C is the conductivity of the hearth material, Btu per (ft, hr, F); D is the diameter or least width of the hearth, in feet; and S is a shape factor equal to 4.00 for circular and 4.40 for square hearths, and varying for rectangular hearths from the last-named value down to 3.73 for very long rectangles.

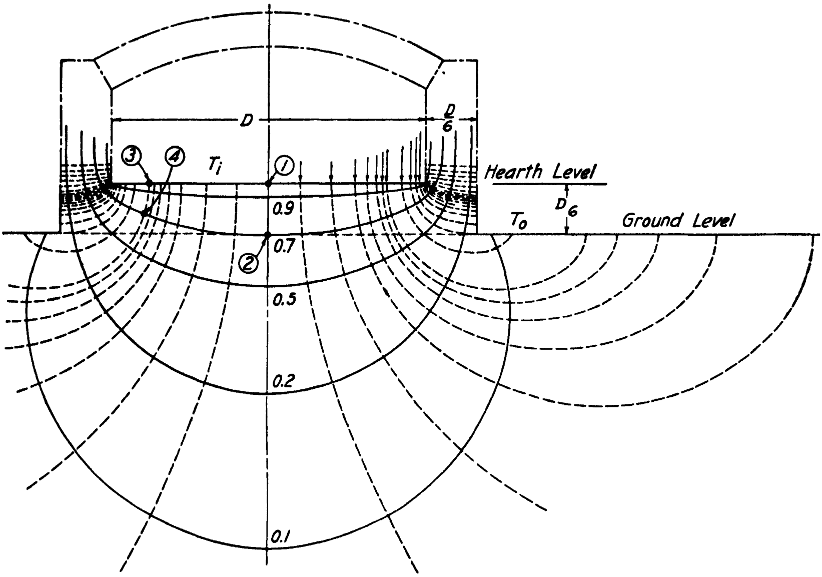

Fig. 89. Heat flow in hearth and foundation. Solid-line curves are isothermals, and numbers on them denote corresponding fraction of total temperature difference $T_i - T_a$.

The loss per unit area is much greater near the edges than near the center; in fact, about one third of the total loss occurs through the outer 5 per cent of the width. But it is in just this section that the greatest change would be produced by changes in the sidewall thickness. The factors just given apply to furnaces having sidewalls of thickness equal to one-sixth the least width of hearth. It is estimated that increasing this thickness to one-fourth the hearth width would reduce the total heat loss 5 per cent, whereas decreasing the sidewalls to one-eighth the hearth width would increase it 8 to 10 per cent.

As a rough average figure, the total hearth loss may be taken as 75 per cent of the loss from an area of sidewall equal to the hearth surface (inside the walls) and of thickness equal to one-sixth the hearth width.

Solid (non-ventilated) hearths are extremely slow in reaching steady-state conditions; they continue to absorb heat to a greater extent than the sidewalls, for weeks and (in large furnaces) months, after lighting up. For further discussion see the paper referred to.

(b) Wall Losses during Intermittent Operations. The heat losses discussed in the preceding section are those incurred during steady, uninterrupted operation. In actual practice, operating periods ("on") alternate with idle periods ("off"). During the off-period, the heat stored in the refractories during the on-period is gradually dissipated, mainly by radiation and convection from the cold face. In addition, some heat is abstracted by air flowing through the furnace. Dissipation of stored heat is a loss, because the lost heat is at least in part again imparted to the refractories during the next on-period.

It is extremely difficult to estimate the heat lost by circulation of air through burner openings and around poorly fitting doors. It is, however, possible to estimate the loss of stored heat that is caused by radiation and convection from the "cold face" while the hot furnace is idle.

Without calculation it is evident that, for short periods of operation between long shutdowns, thin walls of insulating refractories are preferable, because but little heat is stored in them. The heat lost by dissipation of heat stored in the refractories is determined by both the ratio of "on" (operating) time to "off" (idle) time, and the length of the cycle. If a furnace is operated 24 hr every third day, practically all of the heat stored in the refractories is lost. But if the furnace is operated 8 hr per day, not all of the heat stored in the refractories is dissipated.

For the purpose of computing the heat that is lost by intermittent operation of furnaces, several physical properties of refractories must be known. These properties are given in Table 10 and in Fig. 90.

The exact calculation of the heat loss by intermittent operation is troublesome and laborious to such an extent that practice gets along without it. For that reason, an approximate method may be welcome.

Such a method was published by Bradley, Ernst, and Paschkis in

TABLE 10

Kind of Brick	Average Weight, pcf	Specific Heat at Temperature T, or (Mean between 0 F and T F)	Volume Specific Heat = Btu to Raise 1 cu ft of Brick through 1 F, Mean between 0 F and T F	Diffusivity, sq ft per hr
Fireclay brick	120–130	$(0.20 + 0.000035\,T)$	$25 + 0.0044\,T$	0.0204 at 1400 F
Silica brick	105–125	$(0.19 + 0.000039\,T)$	$21 + 0.0043\,T$	0.0262 at 1400 F
Red brick	110 soft burned 130 hard burned	0.200 at 500 F	24	
Silicon carbide	130–155	0.18 at room temp.		0.233 at 2000 F
Magnesite	160–165	$(0.225 + 0.000033\,T)$	$36.5 + 0.0053\,T$	0.0455 at 1400 F
Sillimanite bricks	150–156	0.20 at room temp.	30.6	
Mullite bricks	135–180			
Kaolin bricks	130	$(0.20 + 0.000031\,T)$	$26 + 0.0041\,T$	
Chrome bricks	180–195	$(0.18 + 0.000019\,T)$	$(34 + 0.0036\,T)$	0.023 at 1400 F
Concrete (tamped, 1 : 2 : 5)	150	0.156 at 140 F 0.219 at 770 F		0.0304 at 400 F
Diatomaceous bricks	25–30	0.18 at 1000 F	5.0	0.017 at 700 F
Calcined diatomaceous bricks	36–44	$(0.17 + 0.00003\,T)$	$6.8 + 0.0012\,T$	0.0193 at 1000 F
Lightweight (insulating) firebricks	32–55	Same as fireclay		

T = temperature, F.

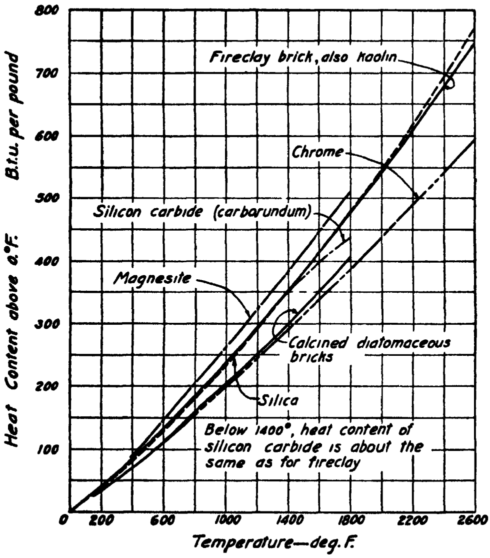

Fig. 90. Heat content of refractory materials.

Transactions American Society of Mechanical Engineers, February, 1946. The data, which were obtained at Columbia University by analogy between flow of heat and flow of electricity, are based on a tight furnace, absence of corner effect, homogeneous wall, constant conductivity of refractory material, infinitely rapid heating of furnace interior, no boundary resistance, and empty furnace. In spite of these severe limitations, the main chart* developed by the three authors is useful. With modifications, the results are here reproduced as Fig. 91. The abscissae represent the ratio of operating time over (operating time plus idle time). The ordinates represent the ratio (wall loss of intermittently operated furnace)/(wall loss of continuously operated furnace). The parameters inscribed on the curves denote the fraction:

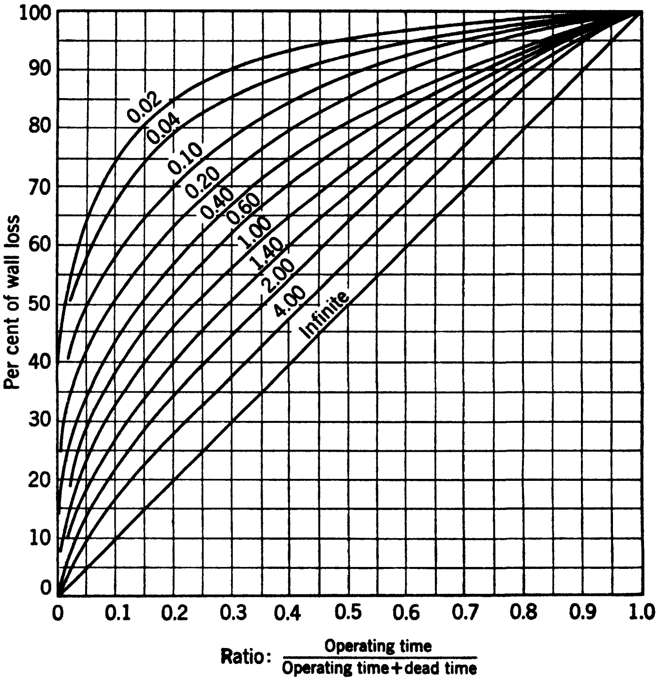

Fig. 91. Wall loss of intermittently operated furnaces in per cent of wall loss of continuously operated furnaces. For meaning of parameter curves see text.

$$\frac{\text{Diffusivity of wall material} \times \text{Length of cycle}}{(\text{Wall thickness})^2}$$

The three coordinates are dimensionless.

At first thought, the ordinates in Fig. 91 seem to conflict with the statement that heat is wasted by intermittent operation. Yet, the indications of less than 100 per cent are correct. If a furnace is fired for 20 hr and remains idle for 80 hr, the wall loss certainly is smaller than it would have been, if the furnace had been fired during 20 + 80 hr.

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* In the original, additional information is given on results obtained by modifying some of the limitations.

Figure 91 does not furnish any information on the wall loss per ton of heated material. A few examples will explain the use of Fig. 91.

EXAMPLE 1. A furnace having firebrick walls 13½ in. thick is operated 8 hr per day of 24 hr. What is the wall loss compared to the loss incurred during steady operation? The abscissa is 8/24 = 0.333. The parameter is

$$\frac{\overset{\text{Diffusivity}}{0.023} \times \overset{\text{Cycle time}}{24}}{(13\frac{1}{2}/12)^2} = 0.44$$

The ordinate belonging to these values is 0.70, or 70 per cent of the steady-state loss (from Fig. 91) for the cycle duration of 24 hours.

EXAMPLE 2. If in example 1, the walls were made of 9-in. refractory insulating brick, what would be the answer?

A typical insulating refractory for high temperature has a density of 50 pcf and a conductivity of 2.5 Btu in. per (sq ft, hr, F). The specific heat is that of firebrick. Then the diffusivity equals $2.5/(12 \times 0.25 \times 50) = 0.17$. The parameter equals $0.17 \times 24/0.75^2 = 0.72$. The corresponding ordinate is 0.64 or 64 per cent.

As previously mentioned, any information gathered from Fig. 91 is only an approximation, not only with regard to heat loss from the cold face, but also because that illustration takes no account of the heat carried off by air flowing through the furnace during the off-period. Electrically heated furnaces are built to be airtight, with the result that practically no air passes through them in the off-period.

A problem that is not strictly part of fuel economy concerns the question: How long a time is required to heat an empty furnace to operating temperature? This problem is briefly mentioned in Chapter 3. Many different factors affect the problem, including the temperature at which the heater judges the furnace to be hot enough; the time elapsed since shutdown; thickness and composition of walls, capacity of burners for releasing heat.

Obviously, it is impracticable to take account of all of these variables. It is, however, feasible to calculate a few simple cases and to use the results for estimating purposes in other cases. A simple case is heating the hot face of an initially cold (room temperature) furnace to 95 per cent of the difference (final wall temperature minus room temperature). The results of such calculations are shown in Fig. 92. The calculations were made by the Schmidt graphical method. (See Appendix 1.) In the illustration, the effects of the type of refractories and of the rate of heat flow into the walls are easily recognized.

In typical operation of industrial furnaces, the refractories lose all of their stored heat only during a prolonged shutdown. At the end of a short idle period much heat is left in the walls. The residual heat may be calculated from Fig. 91. The effect of this residual heat on shortening the heating-up time may be judged by the following method, which furnishes a rough approximation. In example 1, the heat-loss is 70 per cent of the wall loss incurred during steady operation. In the 8-hr firing period, the wall loss is, of course, 100 per cent of the steady-state wall loss, but no heat is abstracted from the heat stored in the refractories. If x be the fraction of the wall loss during

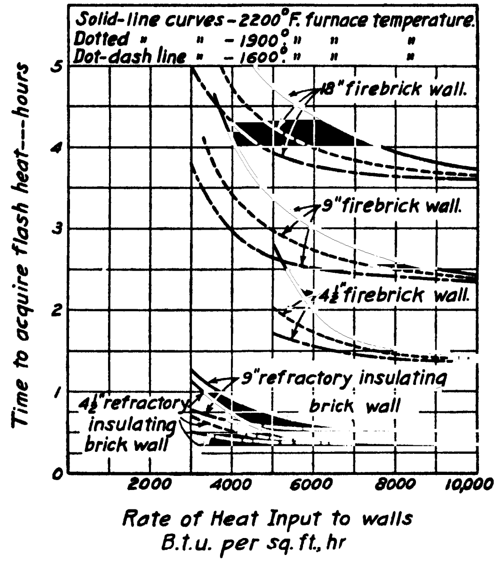

Fig. 92. Calculated time requirement for flash heat. Time for inner surface of furnace wall to complete 95 per cent of that temperature rise at which it would reach the steady state of temperature. If walls retain some heat from previous operation, heating time is reduced.

the 16 hr of idleness, then $8 \times 1 + 16x = 24 \times 0.7$, from which $x = 0.55$. This means that 55 per cent of the heat stored in the refractories is dissipated from the cold face. It also means that only 55 per cent of the heat required for bringing the furnace up to temperature from initially stone-cold conditions need be imparted to the refractories. Because heat is transferred more slowly to a hot surface than to a cold surface, the heating-up time in this example is estimated to be 60 per cent of the time indicated by Fig. 92.

Observation of existing furnaces offers opportunity for measuring the heating-up time of industrial furnaces. Figure 93 was plotted from many observations. The curves refer to initially cold walls and are average values. Figures 92 and 93 were plotted with more and also with less than 5000 Btu per (sq ft, hr) heat input into the walls. This rate of heat input is considered to be an optimum value. With ample capacities of burners, flues, and stack, the time of heating the furnace may be shortened, but the gain of time is small. Moreover, very rapid heating of the walls destroys refractories by spalling.

A detailed example on loss of heat caused by intermittent heating is given at the end of this Chapter (example 1).

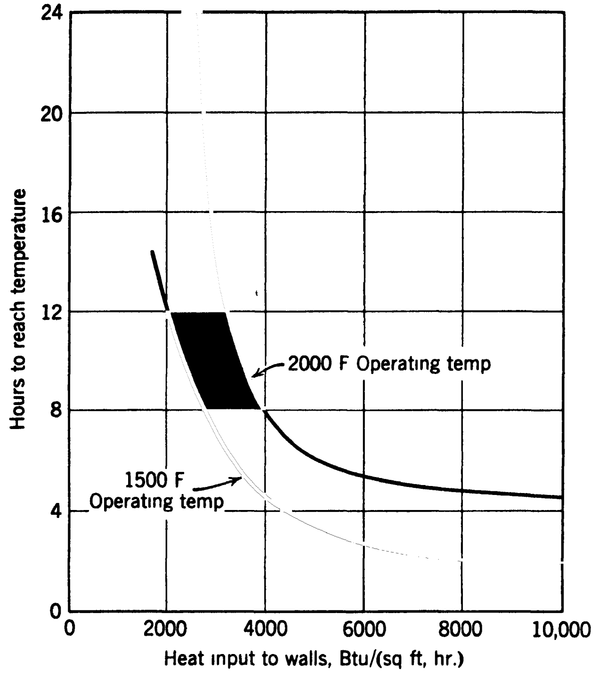

Fig. 93. Time required for heating furnace walls, as observed in practice.

In practice, heating-up times are shorter than indicated by Figs. 92 and 93, because the walls retain much heat during the usually short idle period. The effect of wall thickness is much smaller than indicated by Fig. 92, because the cold air flowing through the furnace during the idle period exerts an equalizing effect which is independent of the wall thickness. With so many overlapping influences, a close theoretical calculation of the heating-up time is almost impossible. For that reason, the following rule of experience may be of value. Furnaces built of insulating refractories heat up in approximately one-half of the time that is required, if the walls are built of dense firebricks.

4. HEAT LOST BY RADIATION THROUGH OPENINGS

Having discussed the heat losses indicated by arrows 2 and 3 in Fig. 77, we can turn our attention to heat losses by direct radiation through openings. These are indicated by arrows 4 of the same illustration.

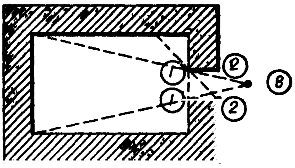

Fig. 94. Diagrammatic illustration of heat radiation from opening.

In Fig. 94, let 1-1 be a hole in the furnace wall. Then the eyes 3 of an observer can see the interior of the furnace, just as if that interior were a plane surface, or "diaphragm," located at 1-1 and having the extent 1-1. If the wall 1-2 were infinitely thin, the diaphragm could be seen from any point to the right of 1-1. The problem would then be reduced to finding the radiation from a freely exposed diaphragm, by the use of equation 6, page 33 or from Fig. 95, which shows the blackbody radiation of a square inch, emitted per hour to surroundings at a temperature of 100 F.

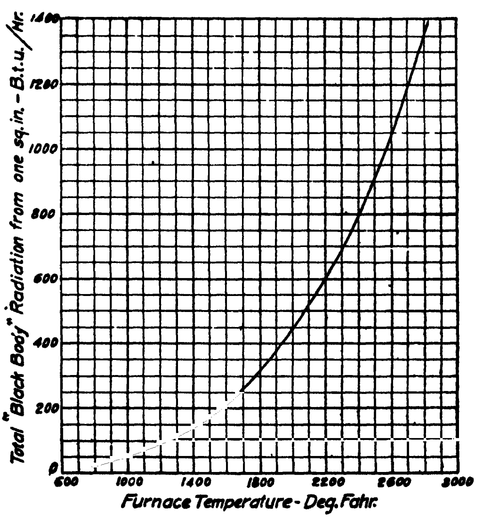

Fig. 95. Blackbody radiation as a function of temperature (not absolute).

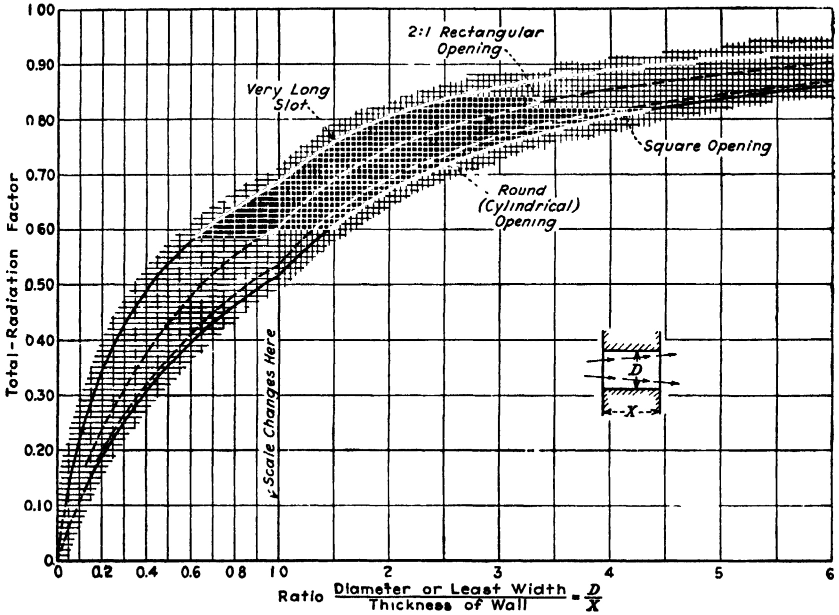

Fig. 96. Radiation through openings of various shapes, as a fraction of the radiation from freely exposed surface of same area as cross section of opening.

The finite thickness of the wall, however, changes matters. It obstructs the direct radiation to an extent depending on the ratio of the wall thickness to the width of opening. On the other hand, the sides of the opening become heated by the radiation that they receive and reradiate a portion of this heat to the outside. The reradiation partially compensates for the reduction of direct radiation. From calculations and tests by J. D. Keller (*Transactions American Society of Mechanical Engineers,* Iron and Steel Division, December 15, 1933, IS–55–6), the ratio of total radiation to the direct radiation from a freely exposed diaphragm has been determined. The values obtained by him are given in Fig. 96 for openings of various shapes.

An example will illustrate the method of calculation. A furnace front is bricked up with a 4½-in. wall, through which an observation opening (peephole) of 2½ by 4½ in. is left open. If the furnace temperature is 2200 F, how much heat escapes through the hole by radiation?

Area of equivalent diaphragm = 2½ in. × 4½ in. = 11.25 sq in. From Fig. 95, blackbody radiation from 1 sq in. of surface at 2200 F = 599 Btu per hr. Heat loss from freely exposed diaphragm = 599 × 11.25 = 6740 Btu per hr.

Ratio of least width of opening to thickness of wall, D/X = 2½ in./4½ in. = 0.556. Ratio of sides of rectangle (equivalent diaphragm) = 4½/2½ = 1.8. In Fig. 96 for $D/X = 0.556$, interpolating between the curves for the 2:1

rectangular opening and the square, the total-radiation factor is found to be 0.45. The actual radiation through opening then is 6740 × 0.45 = 3033 Btu per hr.

The factors apply to steady-state temperature conditions, such as exist when the aperture is open all the time. In door openings, these conditions do not exist. While the door is closed, the sides of the opening (that is to say, the door jambs or piers inside the furnace) become heated practically to furnace temperature from inside to the plane of the door; then, when the door is opened, radiation at first is practically equal to that from a fully exposed diaphragm at the *outside*, as 2-2 (Fig. 94) and can be read directly from Fig. 95. As the door remains open, however, the sides of the opening cool rather quickly, approaching the steady-state temperature distribution and the radiation decreases to the values found by the use of Fig. 96. For further discussion, see the paper referred to.

In (high temperature) furnaces with thick walls, semi-openings can be observed. Unless the binding is very strong, the bricks in the (furnace) interior expand considerably, while the outside layers expand very little. Through the gaps thus produced, the inner, red-hot bricks are visible. The illusion is thus created that the observer can see the furnace interior through the gaps. The inner, red-hot bricks radiate considerable heat through the more or less wide cracks. If the temperature of the inner bricks, the width of the gaps, and the thickness of the outer bricks are known, the heat lost by radiation through the gaps can be computed by Keller's method.

5. HEAT LOST BY FURNACE GASES ESCAPING AROUND DOORS

The loss of heat that is caused by furnace gases passing out around doors, as indicated by arrows 5, in Fig. 77, is particularly hard to estimate. In electric furnaces it is extremely small, unless the doors fit poorly. In combustion-type furnaces, some loss due to this cause invariably occurs, because such furnaces are operated with a slight pressure at the hearth level in the heating chamber. The reason for this method of operation is that a slight loss by gases passing outward is less detrimental than oxidation (and cooling) caused by infiltration of air into the furnace. With tightly fitting doors, the loss need not exceed 2 per cent of the total heat delivered to the furnace and averages 1 per cent as estimated by the authors. With loosely fitting doors, and with a flame or jets of heated gases projected against the doors, it may easily reach 8 per cent, and at times will even exceed that amount.

A distinction must be made between quantity of gases escaping around the doors and available heat carried out by these gases. If combustion has been completed in the furnace, practically no loss is incurred in batch-type furnaces by having gases escape around the doors. It makes no difference whether the gases escape around the doors or pass out through vents into the stack. If, on the other hand, combustion has not been completed, then a heat loss is incurred, because the gases complete their combustion out in the open instead of burning in the furnace. Conditions are different in continuous furnaces. If gases (even if completely burned) pass out through the discharge door, a heat loss is incurred, because these same gases would otherwise give up heat to the charge and would leave the furnace at a lower temperature.

In a continuous furnace with many trouble doors, A. Brandl (*Archiv für das Eisenhüttenwesen,* 1932, page 137) found that 26 per cent of the combustion products containing 3.4 per cent CO and 3.3 per cent H_2, flamed out at the doors. The loss caused thereby was 17 per cent. This furnace was overloaded, however, and worked with excessive pressure. In most continuous furnaces this loss is much smaller.

The loss in question is especially heavy in large forge furnaces with ingots projecting under the door into the open, because large, half-open doors cannot be made tight. In such cases it is better to brick up the opening around the ingot. Since the loss from escaping furnace gases is due partly to the carrying out of sensible heat and partly to carrying out of unconsumed combustible matter, the loss may be counted as part of the flue-gas loss (see arrows 7, Fig. 77).

6. SPECIAL LOSSES

(a) Heat Loss Due to Part of the Stock Projecting out of the Furnace. By reference to Fig. 77, it may be noticed that the next heat loss, in the order given in that illustration, is the loss due to dissipation of heat from metal projecting from the inside of the furnace into the open (Fig. 97). Throughout the heating period, heat flows from the part that is in the furnace to the part that is outside, and a portion of this heat is dissipated from the latter part, by radiation and convection. Any attempt to determine analytically the amount of heat thus conducted and dissipated meets with enormous mathematical difficulties. The differential equations of heat transfer can be written easily enough, but nobody can integrate them, except by introducing so many simplifying assumptions that the result is worthless.* A numerical cut-and-try calculation might be attempted, but it is so everlastingly long that it is never used.

A sufficiently close approximation to the correct solution of the problem can be made by investigating the heat flow for the steady state, i.e., after the temperatures have become practically constant. At that time, just as much heat is dissipated from the cold projection as flows into it from the heated portion in the furnace. During the early part of the heating period, the heat dissipated certainly must be less, because the temperatures are lower. By comparison with tests and by relatively simple reasoning, a sufficiently close approximation can be made to the total quantity of heat lost into and through the projecting part.

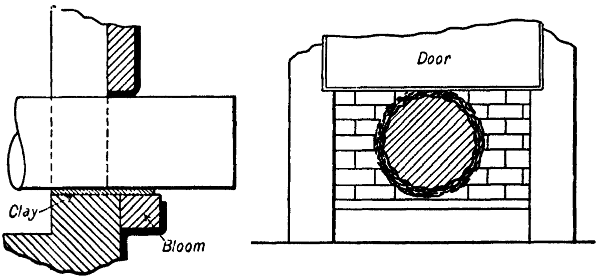

Fig. 97. Ingot projecting out of the furnace.

Figure 98 illustrates the flow of heat and the temperature distribution in the projecting stub of an ingot, after conditions of steady flow have been reached. The curved lines which, in the main, are horizontal, indicate the direction of the flow of heat; the lines that cross the flow lines at right angles are lines of equal temperature. For purposes of calculation, the ingot was divided into a number of concentric layers of equal area. The flow lines dividing the layers are sketched in according to judgment, attention being paid to the temperature distribution. Each of the concentric layers transmits as much heat as the length of the path and the cross-sectional area permit. As the length of the path of heat flow increases from flow line 6 to flow line 1, the temperature gradient drops, and the lowered surface temperature at the outcrop 7 of the flow line results in a lowered heat dissipation.

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* If the variation of temperature across the diameter is neglected, and if the heat-transfer coefficient is assumed to be constant all along the length, then an analytical solution can be obtained, involving hyperbolic sines and cosines, but only for the steady state of temperature. See Ingersoll, Zobel, and Ingersoll, "Heat Conduction," or Gröber, "Wärmeübertragung."

The calculation of the heat loss through a projecting end is extremely tedious, and for that very reason will never be applied in everyday engineering practice; a substitute, therefore, will have to be found. The following reasoning will provide such a substitute.

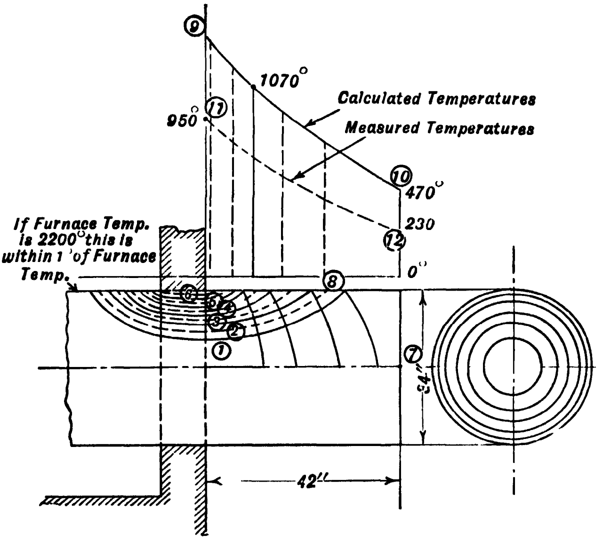

Fig. 98. Heat flow through projecting stub.

The length of the projecting stub is not a quantity of great influence. If the stub is short, the surface is quite hot, but there is not much surface. If the stub is long, there is considerable surface, but its average temperature is low. For that reason, it appears permissible to replace the stub by an equivalent opening from which heat can radiate without obstruction. By comparison with calculations, that equivalent area is found to be equal to 15 per cent of the cross section of the ingot or bar which projects into the open, for usual rates of heating; or 30 per cent when the steady state is reached. If, for instance, a round ingot of 20-in. diameter is being heated, the loss through the stump can be found by taking open-hole radiation through 15 per cent of 314 sq in. or 47.2 sq in. The heat loss through radiation is known from Fig. 95. If, for instance, the furnace temperature equals 2200 F, then the heat loss is 600 Btu per sq in., and 28,500 Btu per hr pass into the projecting stub. It makes no difference whether the heat that passes into the stub stays there or is dissipated; in either case, it is lost, so far as heating the metal in the furnace is concerned.

It will easily be recognized that the method which has just been described is, at best, only a crude approximation. But it is satisfactory, in view of the many uncertainties that surround furnace calculations, such as variable door openings, unknown wall thicknesses (due to burning off or knocking off), and other variables.

(b) Loss by Conduction of Heat through Terminals or Electrodes. Terminals for resistors of various types are shown in Volume II. Although on a smaller scale, the problem is similar to that of heat loss through an ingot stub, but with steady temperature conditions. For the air-cooled terminal, the loss by conduction, Btu per hour, is approximately*

$$Q = n\frac{CA}{l}(T_i - T_a)\frac{l\sqrt{\frac{4k}{CD}}}{1 + l\sqrt{\frac{4k}{CD}}} \tag{18}$$

where n = number of terminals passing through the furnace walls; A = cross-sectional area of terminal, square feet; C = heat conductivity (see page 266), Btu per (ft, hr, F); l = length of terminal passing through wall = thickness of wall, feet; k = heat-transfer coefficient of outer surface of terminal, Btu per (sq ft, hr, F); D = diameter of terminal, feet; T_i = temperature of terminal at inside surface of wall, degrees Fahrenheit; T_a = temperature of atmosphere, degrees Fahrenheit. This loss may be increased if the refractory sleeves around the terminals passing through the wall are quite loose-fitting, in which case heat is radiated through the gap, and hot air flows out carrying heat with it, or if the terminal is so small in cross-sectional area that the heat developed due to its resistance raises the temperature of the part within the wall.

Usually, the leads connected to the terminal are good heat conductors and also have a large area of heat-dissipating surface. In such cases, provided the leads join the terminal relatively close to the furnace wall, equation 18 simplifies to

$$Q = n\frac{CA}{l'}(T_i - T_a) \tag{19}$$

where l' is length from inside of wall to the junction of lead with terminal. If the terminals are of the water-cooled type, the heat loss is much increased, because the intermediate metallic conductor (see illustration in Volume II) must be quite short in order to prevent its overheating and quick deterioration.

* Gröber, "Wärmeübertragung," pages 22–28.

In tall electric furnaces, the loss of heat due to outflow of hot air through the annular spaces between the terminals and the sleeves in the walls through which they pass may be considerable. The sleeves seldom fit closely, and the buoyancy of the column of hot air inside causes outward flow at the top, while cold air is drawn in at the bottom. Sealing the spaces is difficult on account of the electrical insulating requirements.

(c) Heat Imparted to Watercooled Parts. In metal-heating furnaces, skid pipes including supports in continuous furnaces are the only water-cooled parts that are exposed to furnace temperature. For that reason, they are discussed under fuel economy of continuous furnaces. Water-cooled doorframes are not exposed to furnace temperature.

(d) Loss of Heat to Tongs and Charging Machines. Cold tongs absorb heat while in the furnace. Tongs are always cold when they enter the furnace, because they are either water-cooled or else have been in a water trough. Although the flow of heat into tongs is less than the heat that is radiated through the open door, it is by no means inconsiderable and must be taken into account if guarantees of fuel consumption are made.

The heat transmitted to the tongs is the product of (1) exposed surface of tongs, square feet, (2) time of sojourn in furnace, hours, and (3) radiation factor as read from Fig. 95.

The time during which the tongs of the charging machine stay in the furnace is not a fixed quantity, as may be seen from the following observation. The charging machine operator had trouble in separating two heavy billets, which were leaning against each other in the furnace. He kept on trying, until the combination of temperature and force had bent the tongs out of shape. The tongs were then taken out of the furnace, forged back into shape by a sledge hammer, and dipped in water. After that, they were ready for another attempt to remove the obstinate billet.

(e) Loss of Heat from Exposed Surfaces of Lead Baths and Salt Baths. In lead-bath furnaces (Volume II), the loss from the exposed surface of the liquid metal may far exceed the sum of wall losses and useful heat together. Data on radiation constants of molten metals are scarce, but for a bright surface of molten lead the emissivity is apparently about 35 per cent of blackbody emissivity. If the surface is covered with a scum formed by oxidation, the emissivity increases to about 63 per cent. The heat loss would then be given by equation 14, but the constant 0.155 would change to 0.06 for a bright molten surface and to 0.11 for an oxide-covered surface, while the convection factor 0.28 would be increased to about 0.32 (see Appendix, page 458). In practice, for instance in wire-patenting baths, the surface loss is decreased by covering the surface with a layer of crushed or powdered charcoal to a depth of about 1 in. The covering also reduces the loss of metal.

For data on radiation from molten salt baths, see Volume II.

(f) Losses to Trays, Conveyor Chains, and Rollers. The overlooking of this item has caused important furnaces to fail to meet the capacity and fuel consumption guarantees, with disastrous results to the furnace builders who were responsible.

Trays or other containers usually attain the same temperature as the stock, and the amount of heat that they take out of the furnace can be computed from their weight and from the heat-content data on pages 19–20. Chain conveyor parts, on the contrary, especially if thicker than the stock or less freely exposed to heat, do not attain such high temperatures as the stock. Judgment is required in estimating the amount of heat they carry out of the furnace.

Hollow rollers of alloy steel with tapered ends, as described in Volume II, transmit but little heat to the outside. The heat loss may be found from the laws of heat conduction, using the length of the cone for the distance the heat travels and using an average annular cross section as conducting area. The conductivity of alloy steels varies with the temperature.

In furnaces for heating steel to rolling or forging temperatures, conveyor rolls are water-cooled cylinders of mild steel. The heat loss to such rollers is computed in the same manner as the heat lost to other water-cooled parts.

7. SENSIBLE HEAT CARRIED OUT OF FURNACE BY PRODUCTS OF COMBUSTION

In combustion-type furnaces, two additional sources of heat loss occur, namely, those that were indicated by the arrow 7 in Fig. 77. These losses are caused by the heat energy that the products of combustion take out of the furnace, either in potential form in the shape of unburned fuel or in kinetic form in the shape of sensible heat. To sensible heat may be added the latent heat of water vapor.

The chemical composition of the fuel exerts an influence on heat loss (stack loss) here under discussion. At low furnace temperatures, such as 1000 F, the stack loss (heat carried out by products of combustion) is affected but little by the nature of the fuel. At higher temperatures, such as 2400 F, the difference in fuel economy is marked. Figure 76 confirms this statement. The difference is caused by the variation in flame temperatures that can be attained by different fuels. A measure of this maximum temperature is the adiabatic (ideal, theoretical) flame temperature. It is defined as the temperature which is reached if the fuel is burned at constant pressure with room-temperature air and in a heat-tight container. Adiabatic flame temperatures for commonly used fuels, computed without regard to permanent dissociation are shown in Figs. 99 to 108. In furnaces the ideal flame temperatures are never reached, because heat is radiated away while combustion takes place. Nevertheless, a comparison of adiabatic flame temperatures is helpful. If a metal were to be heated to the adiabatic flame temperature, the fuel consumption would be infinitely great and the heating time would be infinitely long.

Calculation of the stack loss is one of the foremost problems of combustion engineering, the solution of which requires tedious and tiresome work. This is, without doubt, one of the reasons why guesswork has been so common in furnace design.

It is, however, quite possible to put the result of these calculations

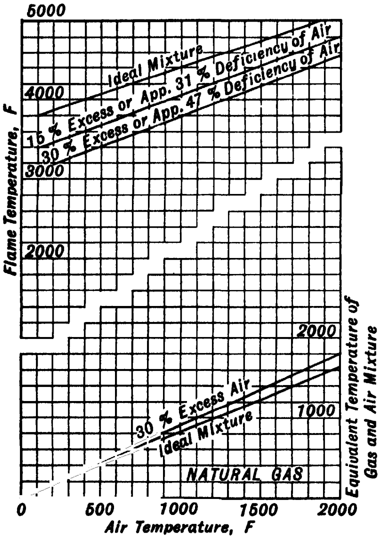

Fig. 99. Adiabatic flame temperatures of natural gas, with air preheated to various temperatures.

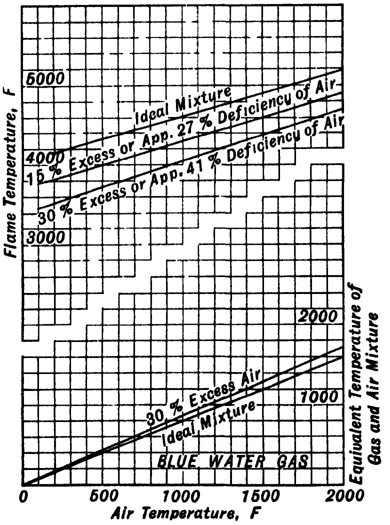

Fig. 100. Adiabatic flame temperatures of blue water gas, with air preheated to various temperatures.

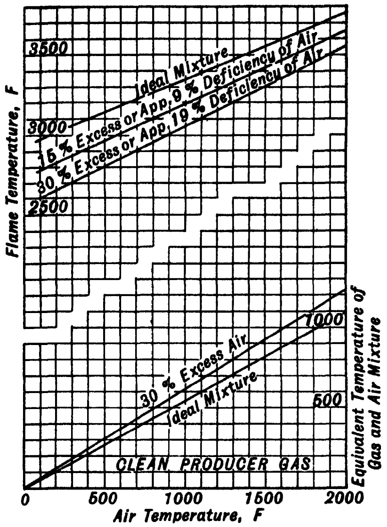

Fig. 101. Adiabatic flame temperatures of clean producer gas, with air preheated to various temperatures.

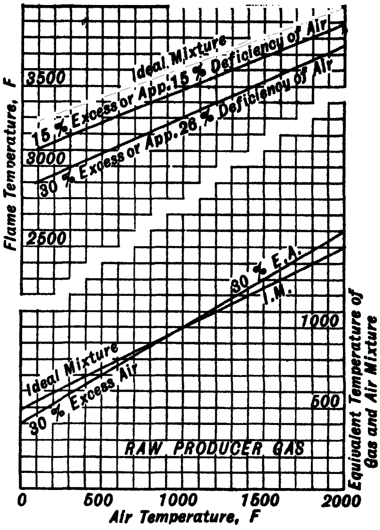

Fig. 102. Adiabatic flame temperatures of raw producer gas, gas initially at 1000 F; air preheated to various temperatures.

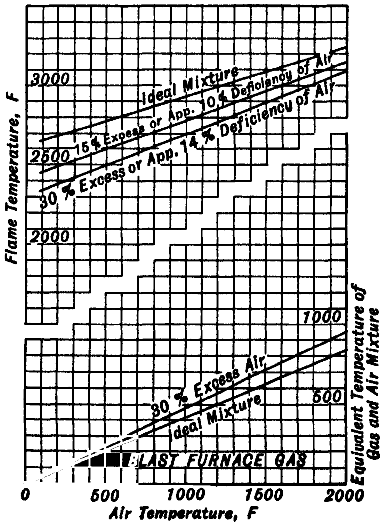

Fig. 103. Adiabatic flame temperatures of blast-furnace gas, with air preheated to various temperatures.

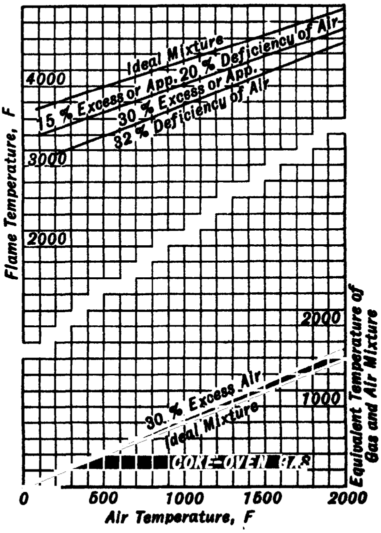

Fig. 104. Adiabatic flame temperatures of coke-oven gas, with air preheated to various temperatures.

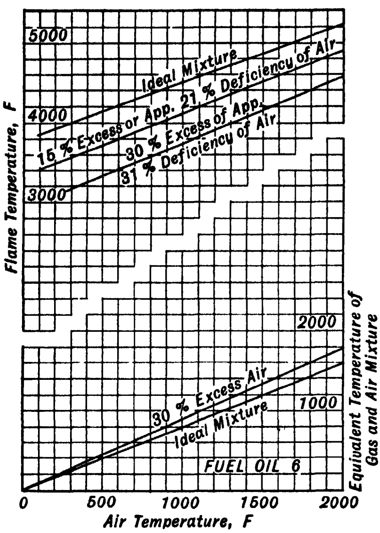

Fig. 105. Adiabatic flame temperatures of fuel oil 6, with air preheated to various temperatures.

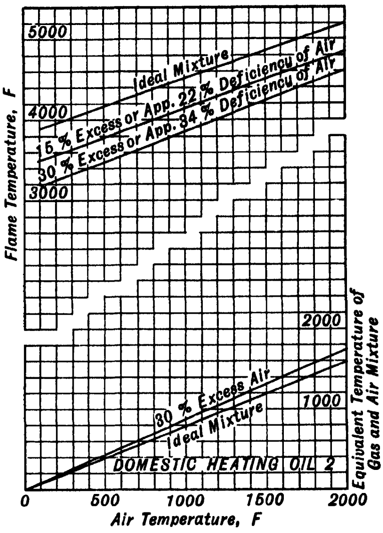

Fig. 106. Adiabatic flame temperatures of fuel oil 2, with air preheated to various temperatures.

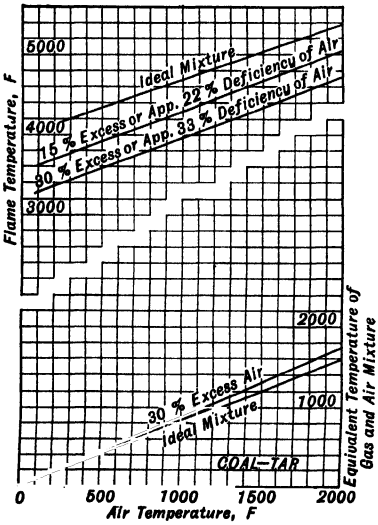

Fig. 107. Adiabatic flame temperatures of raw coal tar, with air preheated to various temperatures.

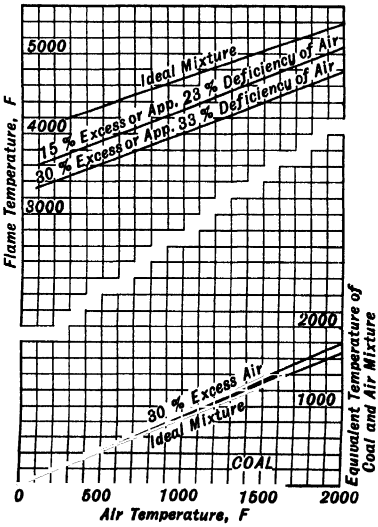

Fig. 108. Adiabatic flame temperatures of coal, with air preheated to various temperatures.

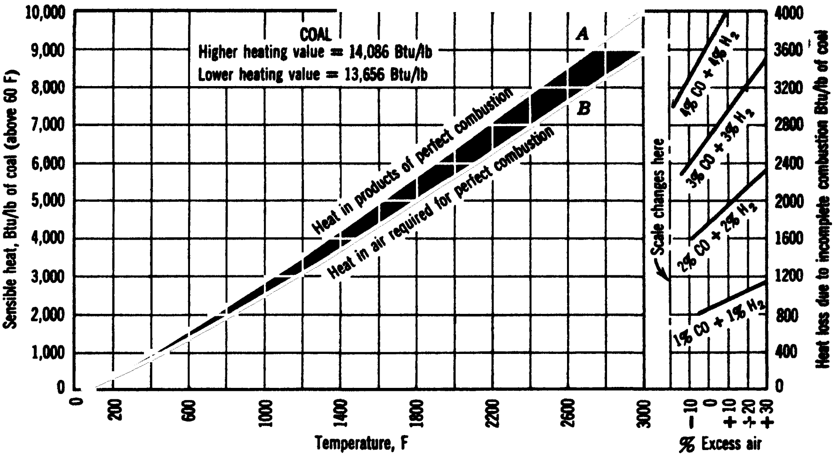

Fig. 109. Heat in products of combustion of high-grade bituminous coal.

Ultimate analysis of coal:

C	79.86%
H	5.02
O	4.27
N	1.86
S	1.18
Ash	7.81
	100.00%

Air required for perfect combustion = 10.8 lb or 141.5 cu ft per lb of coal.

Gaseous products of perfect combustion:

	Lb per lb of coal
CO_2	2.953
H_2O	0.452
N_2	8.318
Total	11.723

Specific gravity of gaseous products referred to air as unity = 1.042.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 430 Btu to sensible heat content.

If no hydrogen is present in the complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

into graphical form, once for all, and to use the charts afterwards with but little trouble, without loss of time, and with but few chances for errors. To this end, the heat content of the flue gases is plotted against their temperature, as indicated in Fig. 109 and the figures that follow. The *A* curve in those illustrations furnishes the sensible heat content (above 60 F) in the products of perfect combustion (without excess air) of unit quantity of fuel, that unit depending on the form of fuel. It does not include the latent heat of the water vapor. For this reason, the heating value that should be used in calculations with the curve is the lower heating value. If the higher heating value is used instead, then the latent heat of the water vapor

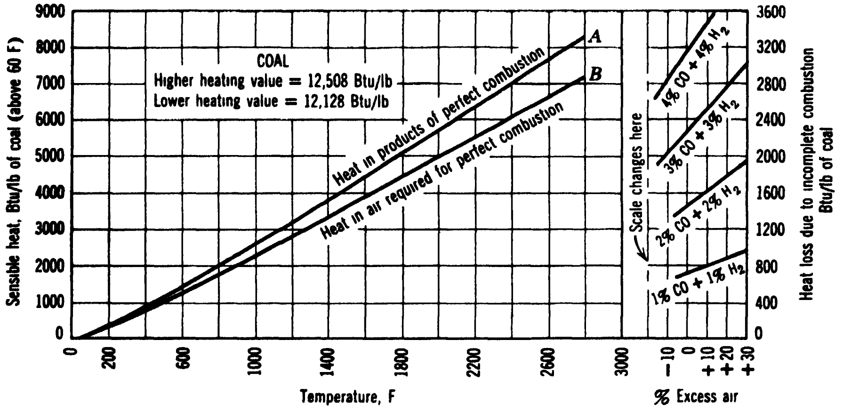

Fig. 110. Heat in products of combustion of low-grade bituminous coal.

Ultimate analysis of coal:

C	70.00%
H	5.00
O	8.00
N	2.00
S	2.00
Ash	13.00
	100.00%

Air required for perfect combustion = 9.53 lb or 125 cu ft per lb of coal.

Gaseous products of perfect combustion:

	Lb per lb of coal
CO_2	2.61
H_2O	0.45
N_2	7.34
Total	10.40

Specific gravity of gaseous products referred to air as unity = 1.039.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 380 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

(which is the difference between the higher and the lower heating value) should be added to the heat content as read from the curve *A*.

The *B* curve, taken by itself, has no direct physical meaning, but it is very convenient for purposes of calculation. It represents the heat in that quantity of air which is required to burn a unit quantity of fuel, plotted as a function of the temperature of that air. If combustion has been carried on with excess air, or if air has filtered into the furnace after combustion, the flue gases contain excess air. If the amount of that excess air is *p* per cent, the heat in the flue gas per unit quantity of fuel, and for a given temperature, is found by taking the ordinate of the *A* curve for the temperature in question and adding to it *p* per cent of the ordinate of the *B* curve. Examples for the use of the curves will be found below. Before these examples are given, several other points

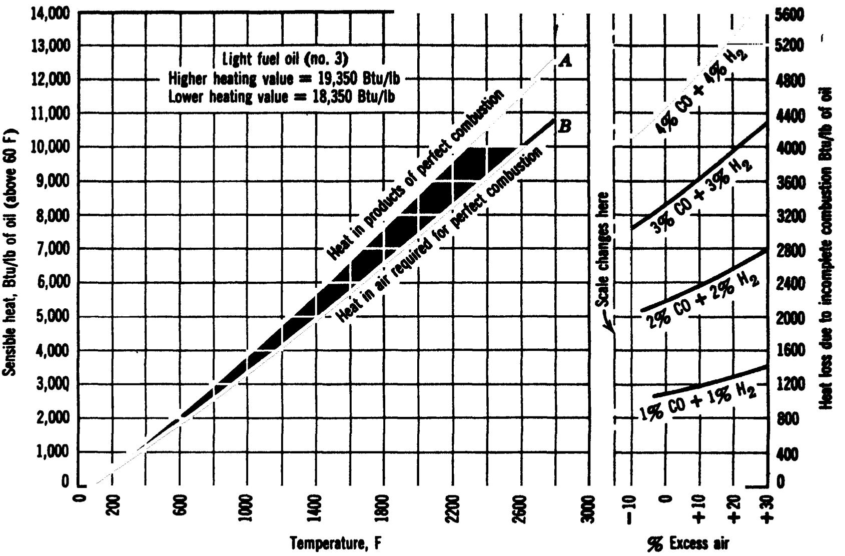

Fig. 111. Heat in products of combustion of light fuel oil.

Lb of oil per gal = 7.39
Density, degrees API = 28
Ultimate analysis:

C	86.8%
S	0.7
H	12.3
N + O	
Plus ash	0.2
	100.0%

Air required for perfect combustion = 14.24 lb per lb of oil, or 1377 cu ft per gal (at 60 F).

Products of perfect combustion:

	Lb per lb of oil
$CO_2 + SO_2$	3.202
H_2O	1.100
N_2	10.938
Total	15.240

Specific gravity of products referred to air as unity = 1.004.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 1155 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

about the heat-content curves must be taken up. An inspection of the curves for different fuels reveals the fact that in some of the curve sheets a third curve *C* has been entered. This latter curve shows the sensible heat in unit quantity of fuel. In later calculations on furnace economy, the *C* curve will be found very useful for fuels that enter the furnace in a preheated state.

The heat-content curves are strictly correct for those fuels only that have the compositions given in connection with the charts. In prac-

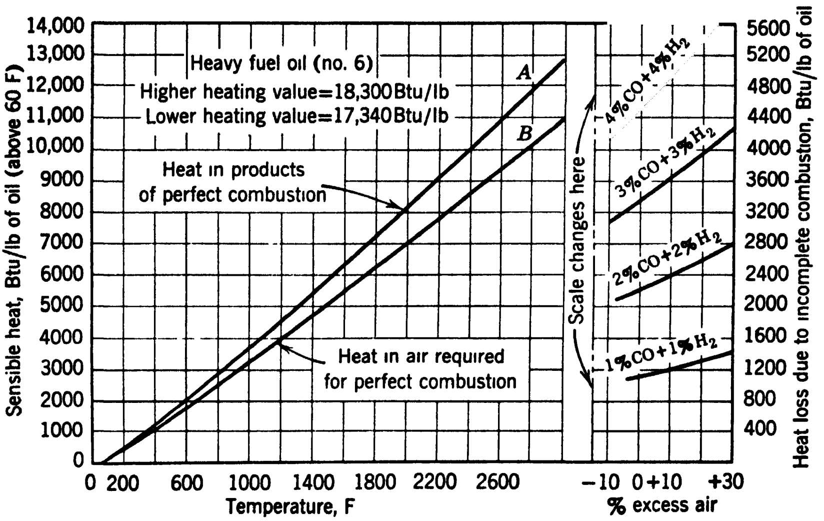

Fig. 112. Heat in products of combustion of heavy fuel oil.

Lb of oil per gal = 8.27
Density, degrees API = 11
Ultimate analysis:

C	86.8%
H	10.2
S	2.0
N + O	
Plus ash	1.0
	100.0%

Air required for perfect combustion = 13.57 lb per lb of oil, or 1468 cu ft per gal (referring to 60 F).

Products of perfect combustion:

	Lb per lb of oil
$CO_2 + SO_2$	3.228
H_2O	0.911
N_2	10.431
Total	14.570

Specific gravity of products referred to air as unity = 1.012.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 960 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

tice, fuels bearing the same name but having somewhat different compositions are encountered. Thus, the composition of natural gas varies with the locality of the origin of the gas. The composition of by-product coke-oven gas varies with the composition of the coking coal, the coking time, and the tightness of the ovens. With very few exceptions, the deviations are so small that no serious error is introduced by using the heat-content curves (Figs. 109 to 121). If many calculations are to be made for a widely different fuel, such as, for instance, refinery gas, then a new chart can be made by following the method

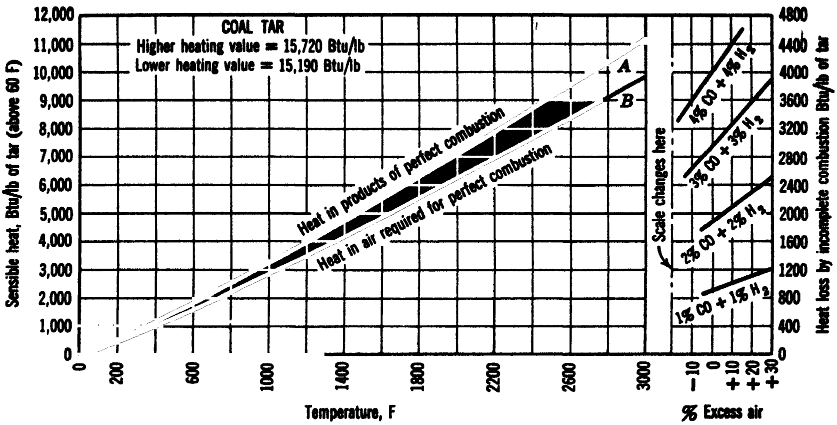

Fig. 113. Heat in products of combustion of coal tar (not topped).

Lb of tar per gal	9.49
Specific gravity	1.139
Flash point	172° F.

Ultimate analysis:

C	86.70%
H	6.00
O	3.10
N	0.15
S	0.75
Ash	0.10
Water	3.20
	100.00%

Air required for perfect combustion = 12.052 lb or 158 cu ft per lb of tar.

Products of perfect combustion:

	Lb per lb of tar
CO_2	3.180
H_2O	0.572
N_2	9.300
Total	13.052

Specific gravity of products referred to air as unity = 1.034.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 530 Btu to sensible heat content.

If no hydrogen is present in the complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

that is explained in the example given below. The short curves on the right of the charts are explained later.

The method of calculation which forms the basis for the curve sheets, Figs. 109 to 121, will now be given for the benefit of those who may have occasion to do similar work for a fuel not included in the curve sheets of this chapter.

The calculation is made for the products of combustion of light fuel oil (Fig. 111) and for 2000 F temperature of products of combustion. By the usual methods of calculating chemical reactions, the weight of oxygen needed for combustion and the weight of nitrogen that accompanies the oxygen are found.

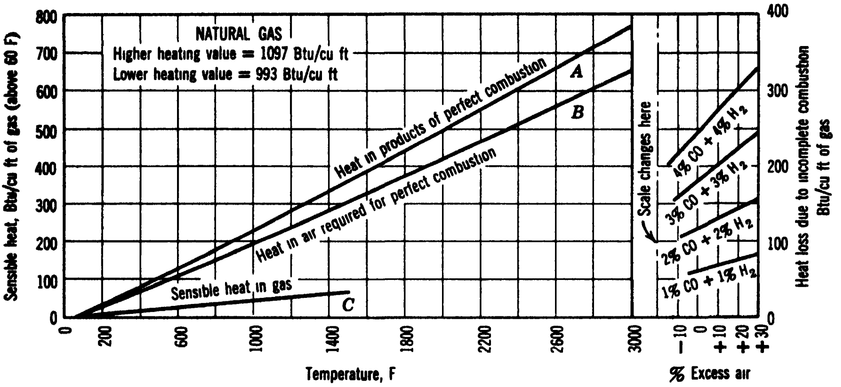

Fig. 114. Heat in products of combustion of natural gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of gas = 0.0486 pcf.

Analysis of gas by volume:

CH_4	87.0%
C_2H_6	7.6
C_3H_8	2.7
C_4H_{10}	0.8
N_2	1.9
	100.0%

Air required for perfect combustion = 0.8037 lb or 10.5 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
CO_2	0.1318
H_2O	0.1005
N_2	0.6140
Total	0.8463

Specific gravity of products referred to air as unity = 0.955.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 104 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

The products of combustion of 1 lb of fuel are:

$$CO_2 + SO_2 = 3.202 \text{ lb}$$

$$H_2O = 1.100 \text{ lb}$$

$$N_2 = 10.938 \text{ lb}$$

The increase of heat contents from zero to 2000 F is either read from charts of heat contents or is calculated by use of mean specific heats. If neither is available, the mean specific heat may be found by analytical or numerical integration over the desired temperature range. By analytical integration, the mean specific heats at constant pressure are: CO_2, 0.273; N_2, 0.271; H_2O, 0.517; air, 0.265. With these values, heat in products of combustion above zero equals

$$(3.202 \times 0.273 + 10.938 \times 0.271 + 1.100 \times 0.517) \times 2000$$
$$= 8814 \text{ Btu per lb of fuel.}$$

From zero to 60 F, the mean specific heats are: CO_2, 0.205; N_2, 0.247; H_2O, 0.445; air, 0.242. Heat in products of combustion not condensed from 1 lb of fuel between 60 and 2000 F equals 8814 − 230 = 8584 Btu. This does not include the latent heat of water vapor. By the use of the mean specific heats for air, the heat required to raise the temperature of 14.24 lb of air from 60 to 2000 F equals 7330 Btu. This latter figure furnishes the 2000 F point on the *B* curve.

The use of the curves (Figs. 109 to 121) is illustrated by the following example after Fig. 116 on page 154.

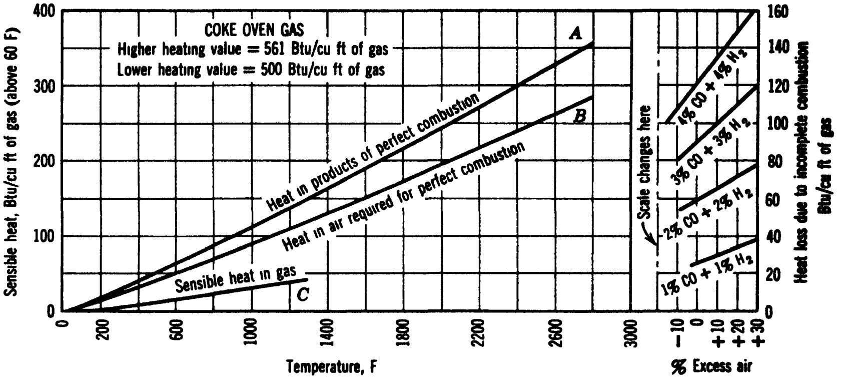

Fig. 115. Heat in products of combustion of by-product coke-oven gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of gas (saturated) = 0.0313 pcf.

Analysis of gas by volume:

H_2	47.3%
CH_4	34.2
CO	6.9
C_2H_4	2.6
CO_2	2.2
O_2	0.3
N_2	6.0
H_2S	0.5
	100.0%

Air required for perfect combustion = 0.380 lb or 4.97 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
$CO_2 + SO_2$	0.057
H_2O	0.061
N_2	0.293
Total	0.411

Specific gravity of products referred to air as unity = 0.938.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 61 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

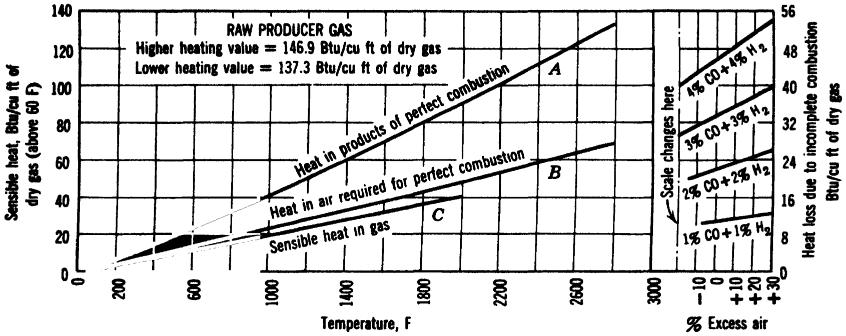

Fig. 116. Heat in products of combustion of raw producer gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of dry gas = 0.0680 pcf.

Equivalent density referred to 60 F of the gas including water vapor = 0.0668 pcf.

Analysis of (dry) gas by volume:

CO	20.5%
H_2	12.5
CH_4	3.0
CO_2	7.5
N_2	56.5
	100.0%

In addition, each cubic foot of dry gas contains 0.000625 lb of tar vapors and 0.0030 lb of water vapor.

Air required for perfect combustion = 0.0900 lb or 1.178 cu ft per cu ft of dry gas.

Products of perfect combustion:

	Pcf of dry gas
CO_2	0.0382
H_2O	0.0124
N_2	0.1110
Total	0.1616

Specific gravity of products of combustion referred to air as unity = 1.01.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 9.6 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

Example. Let natural gas and air enter a furnace at atmospheric temperature; let the combustion take place with 10 per cent excess air; and let the products of combustion leave the furnace with a temperature of 2300 F. What fraction of the heat of combustion stays in the furnace, and what fraction is taken away by the products of combustion?

In Fig. 114, the heat in the products of perfect combustion is found to be 575 Btu at 2300 F. The heat in the air is found to be 480 Btu at 2300 F, which means that 10 per cent excess air contains 48 Btu. The heat in the flue gases = 575 + 48 = 623 Btu per cu ft of fuel gas. The higher heat value of the fuel gas equals 1097 Btu per cu ft; its lower heat value, 993 Btu per cu ft. Hence, 623 Btu are carried out as sensible heat in the products of combustion, and 1097 − 993 = 104 Btu as latent heat in the water vapor produced. Thus, (623 + 104)/1097 = 0.66, or 66 per cent of the total heat, is

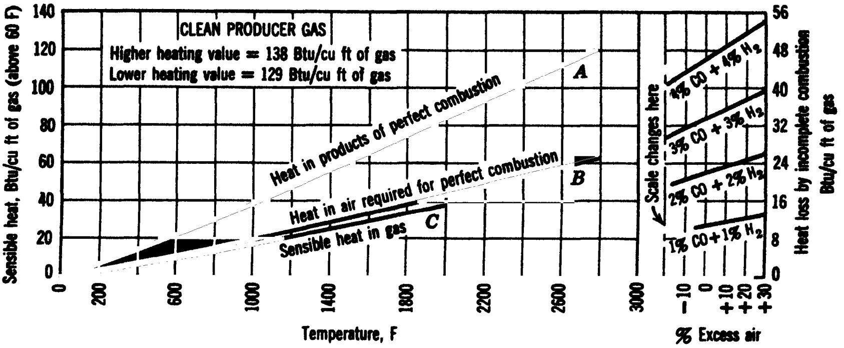

Fig. 117. Heat in products of combustion of clean producer gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of gas = 0.0679 pcf.

Analysis of gas by volume:

CO	19.03%
H_2	13.48
CH_4	2.78
C_2H_4	0.19
O_2	0.02
CO_2	9.71
N_2	54.79
	100.00%

Air required for perfect combustion = 0.0814 lb or 1.065 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
CO_2	0.0371
H_2O	0.0092
N_2	0.1030
Total	0.1493

Specific gravity of products referred to air as unity = 1.020.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 9 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

carried off by the products of combustion, and the rest, namely, 34 per cent, is given up in the furnace.

Under each of the illustrations (Figs. 109 to 121) the composition of the products of perfect combustion is given on a weight basis. The following calculation, using Fig. 118 as an example, converts the composition to a volume basis. The weight constituents are divided by their respective molecular weights, whereby volume parts are obtained. Each volume part is divided by the sum of the volume parts and is multiplied by 100, in order to obtain the composition in per cents by volume.

$$\left.\begin{array}{lll} CO_2 & 0.0792/44 & = 0.00180 \\ H_2O & 0.0473/18 & = 0.00263 \\ N_2 & \underline{0.2787/28} & = \underline{0.00995} \end{array}\right\} \text{Volume parts}$$

$$\begin{array}{ll} \text{Sum} & 0.01438 \end{array}$$

$$100 \times 0.00180/0.01438 = 12.5\%\ CO_2$$

$$100 \times 0.00263/0.01438 = 18.3\%\ H_2O$$

$$100 \times 0.00995/0.01438 = 69.2\%\ N_2$$

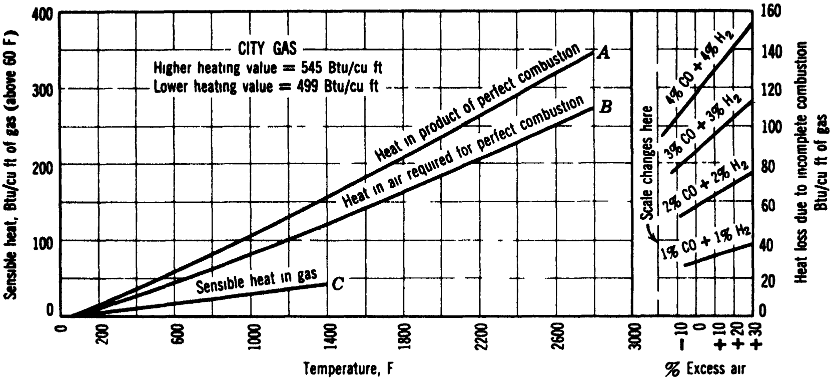

Fig. 118. Heat in products of combustion of a representative city gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of gas (saturated) = 0 0471 pcf.

Analysis of dry gas by volume:

H_2	36.8 %
CO	23.9
CH_4	17.8
C_2H_4	5.34
C_3H_6	1.46
C_4H_8	0.94
C_6H_6	0.86
CO_2	3.7
O_2	1.2
N_2	8.0
	100.00%

Air required for perfect combustion = 0.3581 lb or 4.68 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
CO_2	0.0792
H_2O	0.0473
N_2	0.2787
Total	0.4052

Specific gravity of products referred to air as unity = 0 972.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 46 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

If an Orsat analysis (dry basis) is wanted, CO_2 and N_2 only are added, and their sum is called 100 per cent.

8. HEAT LOSS BY INCOMPLETE COMBUSTION

Unconsumed fuel passes out of many industrial furnaces. In order to reduce oxidation of the charge, heaters sometimes keep the furnace atmosphere smoky. With the theoretically correct fuel to air ratio, combustion is not completed unless fuel and air are thoroughly mixed.

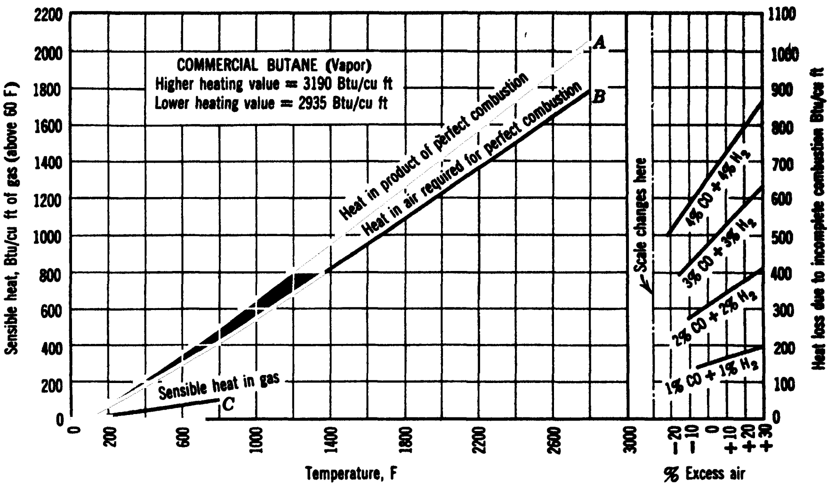

Fig 119. Heat in products of combustion of commercial butane.

Lb of liquid per gal = 4.8.
Specific gravity = 0.577.
Ultimate analysis:

C_4H_{10}	93.0%
C_3H_8	7.0
	100.0%

Density of vapor = 0.150 pcf at 60 F, 14.7 psi abs pressure.

Higher heating value = 21,300 Btu per lb or 102,200 Btu per gal.

1 gal of liquid produces 32 cu ft of vapor.

Air required for perfect combustion = 2.34 pcf of gas or 30.8 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
CO_2	0.4563
H_2O	0.2406
N_2	1.7861
Total	2.4830

Specific gravity of products referred to air as unity = 0.982.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 255 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

Even with excess air, combustible is occasionally found in the waste gases. The amount of heat lost in this way, therefore, depends not only on the design of the burners and furnace but also on operating requirements.

Industrial fuels, with the exception of coke and blast furnace gas, contain hydrocarbons. In the initial stages of combustion, compounds such as CH_nOH are formed. With excess air, these compounds are oxidized to CO_2 and H_2O. With insufficient air supply (in its entirety or locally), the end products are CO_2, CO, H_2O, H_2, and methane. If

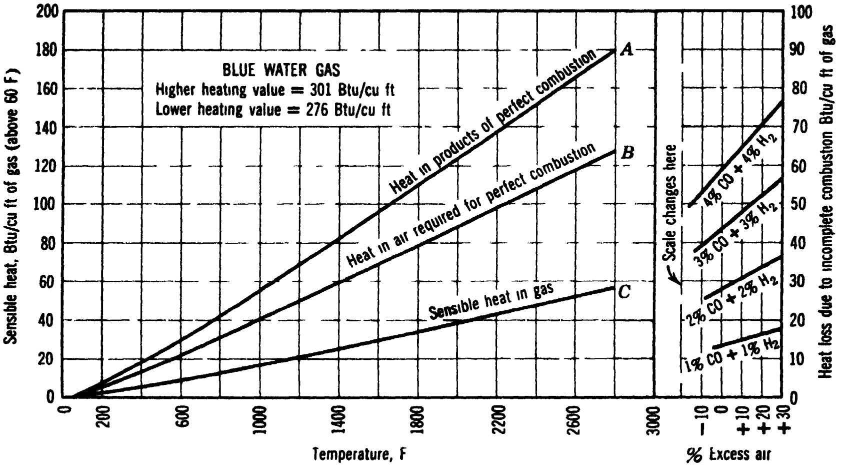

Fig. 120. Heat in products of combustion of blue water gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of gas = 0.0427 pcf.

Analysis of gas by volume:

CO	43.5%
H_2	47.3
CH_4	0.7
O_2	0.6
CO_2	3.5
N_2	4.4
	100.0%

Air required for perfect combustion = 0.1695 lb or 2.23 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
CO_2	0.05527
H_2O	0.02328
N_2	0.13365
Total	0.21220

Specific gravity of products referred to air as unity = 0.989.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 25 Btu to sensible heat content.

If no hydrogen is present in a complete analysis of the flue gases, multiply incomplete combustion loss on scale at right by 51 per cent.

the products of partial combustion are chilled, soot is formed.

Several experimenters* have found that, when CO is present, it is accompanied by an approximately equal amount of H_2, or H_2 plus CH_4. The CH_4 appears at low furnace temperatures when much H_2 is present.

Because the loss caused by incomplete combustion depends on equal volumes of CO and H_2, the loss caused by CO must be determined. The percentage of CO in the flue gas is not a direct measure of the loss due to incomplete combustion. Instead, based on the fact that

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* Pistor, *Archiv für das Eisenhüttenwesen,* 1931, page 135; B. Houghton, *Proceedings Third International Conference on Bituminous Coal,* 1931, Volume 2, page 279; also private communication, Trinks and Keller (not published).

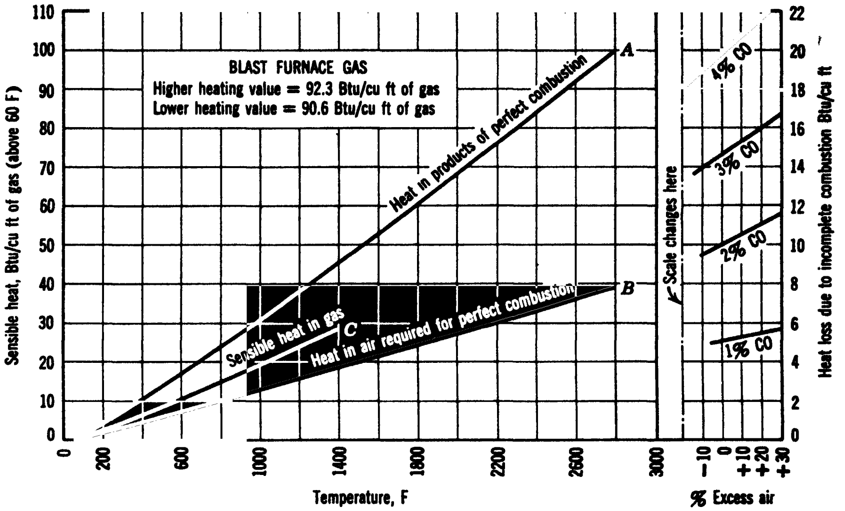

Fig. 121. Heat in products of combustion of blast-furnace gas.

All volumes are at 60 F and 14.7 psi abs pressure.

Density of gas = 0.0768 pcf.

Analysis of gas by volume:

CO	25.40%
H_2	3.50
CO_2	12.50
N_2	58.60
	100.00%

Air required for perfect combustion = 0.0529 lb or 0.695 cu ft per cu ft of gas.

Products of perfect combustion:

	Pcf of gas
CO_2	0.0442
H_2O	0.0017
N_2	0.0838
Total	0.1297

Specific gravity of products referred to air as unity = 1.094.

Curve *A* does not include latent heat of water vapor. Use lower heating value or, if higher heating value is used, add 1.7 Btu to sensible heat content.

a given weight of carbon makes the same volume, whether burned to CO or CO_2, and that both gases are measured in the flue-gas analysis, the following expression for the loss due to incomplete combustion of carbon is derived.*

$$L = 10{,}150\,C \times \frac{\%\,CO}{\%\,CO + \%\,CO_2} \tag{20}$$

where 10,150 = difference in heat evolved between burning 1 lb of carbon to CO_2 and to CO, and where C = fraction (by weight) of carbon in the fuel which is burned. If as much H_2 as CO escapes unburned, the additional loss, based on the lower heating value (282 Btu per cu ft compared with 322 Btu per cu ft for CO), is 87 per cent of the above. The total loss due to incomplete combustion then is

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* The per cent CO_2 is that derived *from combustion only*. If some CO_2 is initially present, as in blast-furnace gas, a modification is required; it has been taken care of in Table 11 and in Figs. 115 to 118, 120, and 121.

$$L = 19{,}100\,C \times \frac{\%\,\mathrm{CO}}{\%\,\mathrm{CO} + \%\,\mathrm{CO_2}} \qquad (21)$$

Table 11 gives an idea of the loss due to incomplete combustion for different fuels.

TABLE 11

	Heat Loss Indicated by Presence in Dry Flue Gases (When There Is no Excess or Deficiency of Air) of	
Fuel	1% CO + 1% H_2 Per Cent of Higher Heat Value	1% CO, no H_2 Per Cent of Higher Heat Value
Coal, 70% carbon	*5.94	*3.17
Coal, 80% carbon	*6.00	*3.20
Oil	5.53	....
Natural gas	5.02	....
By-product coke-oven gas	4.78	....
Producer gas (either raw or clean)	7.50	....
Blue water gas	4.60	....
Coke	....	3.42
Blast-furnace gas	....	5.17

* Loss varies, depending on amount of hydrocarbons present; see page 157.

A still better grasp of the losses due to incomplete combustion can be obtained by consulting Figs. 109 to 121, in which these losses have been entered at the right-hand side. A glance at any of these illustrations shows that a given percentage of CO in the stack gases means an increasingly greater loss if more and more excess air is used in combustion; most industrial furnaces, however, work with a deficiency of air. It is, of course, obvious that the simultaneous appearance of much CO and of O_2 in the stack gases is the result of extremely poor mixing of the fuel and air.

9. OSTWALD CHARTS

In order to compute the excess or deficiency of air from a flue-gas analysis, one must know the ultimate analysis of the fuel and must be familiar with combustion calculations. These computations may be avoided by the use of graphical charts such as were invented by

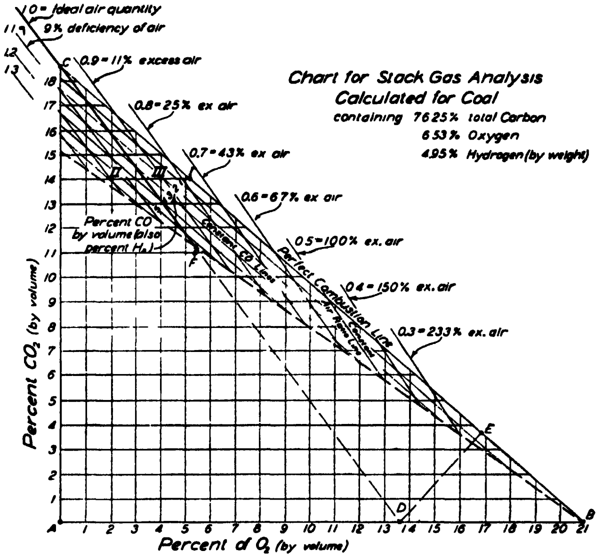

Fig. 122. Ostwald chart for coal. Based on H_2 in products being equal to CO in volume per cent and on no soot being formed.

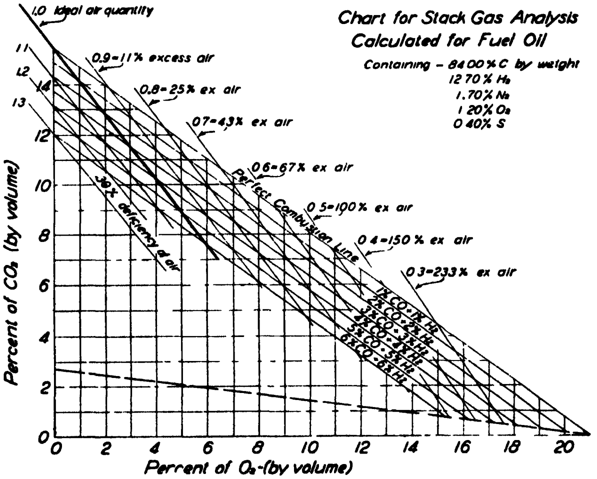

Fig. 123. Ostwald chart for fuel oil. Based on H_2 in products being equal to CO in volume per cent and on no soot being formed.

W. Ostwald. In the original Ostwald charts, however, the assumption was made that all H_2 is burned and that the only combustible gas in the stack gases is CO. The charts were modified by J. D. Keller to take account of the experimentally determined fact, mentioned on page 158, that when CO appears, an equal volume of H_2 is usually present. Figures 122 to 127 are the modified Ostwald charts for different fuels.* They cover a sufficiently wide range to allow a very close estimate of the excess air from a dry-gas analysis and from an approximate knowledge of the nature of the fuel.

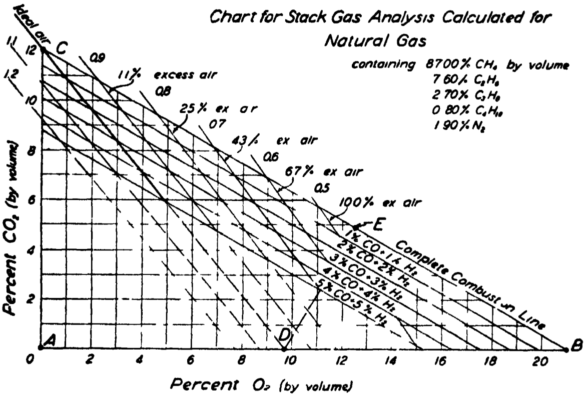

Fig. 124. Ostwald chart for natural gas. Based on H_2 in products being equal to CO in volume per cent and on no soot being formed.

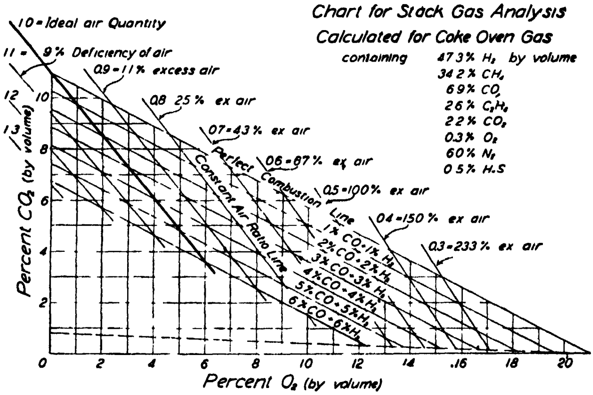

Fig. 125. Ostwald chart for coke-oven gas. Based on H_2 in products being equal to CO in volume per cent and on no soot being formed.

The Ostwald charts are inaccurate, if soot is one of the products of incomplete combustion.

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* The method of calculating and constructing the modified charts is described in *Blast Furnace and Steel Plant,* September and October 1932.

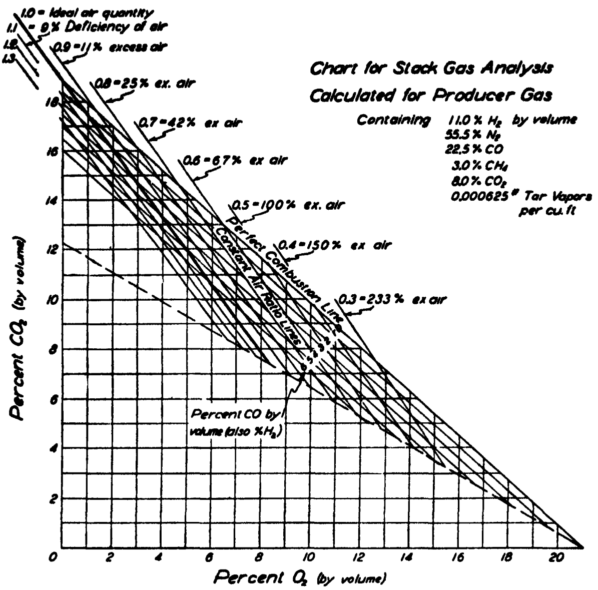


Fig. 126. Ostwald chart for producer gas. Based on H_2 in products being equal to CO in volume per cent and on no soot being formed.

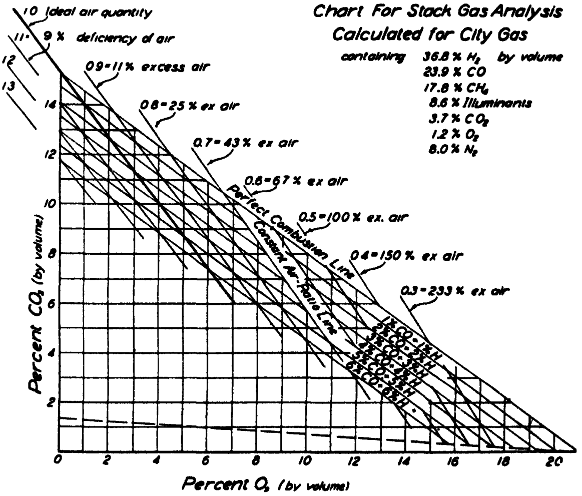


Fig. 127. Ostwald chart for a representative city gas. Based on H_2 in products being equal to CO in volume per cent and on no soot being formed.

Examples will illustrate the use of the charts. With bituminous coal as fuel (Fig. 122), a stack-gas analysis shows $CO_2 = 14.0\%$ and $O_2 = 5.2\%$. By locating 14 per cent CO_2 on the left-hand vertical edge, *AC*, and 5.2 per cent O_2 on the base line, *AB*, and by following horizontal and vertical lines denoting these two quantities to their intersection at point I, it is found that the representative point on the chart lies on the line of perfect combustion, so that there is no CO in the products of combustion. Air ratio, as found from the slightly inclined lines, equals 0.76; its reciprocal equals 1.32, which means that combustion was carried on with 32 per cent excess air.

If the gas analysis had shown 14 per cent CO_2, as before, but only 2.0 per cent O_2, then point II of Fig. 122 would have been reached. This point lies 5.9 inclined spaces away from the perfect combustion line, *CB,* which means that the gases contain 5.9 per cent CO. If the analysis shows a different amount, the probabilities are that an error has been made in it. The air ratio (as determined by the location of point II with regard to the slightly inclined air-ratio lines) is found to be 1.20; its reciprocal is 0.833, which means that combustion took place with 16.7 per cent air deficiency.

If, finally, the stack-gas analysis had shown 14 per cent CO_2 and 4 per cent O_2, then point III would be reached on the chart. It is located 2.2 CO spaces distant from the line of perfect combustion; accordingly, the stack gases must contain 2.2 per cent CO, and, on the basis previously described, the hydrogen is also 2.2 per cent. The air ratio (slightly inclined lines) is 0.91; its reciprocal is 1.10, which means that combustion took place with 10 per cent excess air.

For other fuels, information is gathered from the corresponding charts in the same manner. To construct charts for fuels of different compositions from those given, the methods described in the reference given above should be used.

In the design of new furnaces, the incomplete-combustion loss is best estimated on the basis of results from previously constructed furnaces. The amount of combustible escaping unconsumed depends not only on the amount of air supplied, the burner design (thoroughness of mixing versus stratification), the furnace temperature, the air preheat, and the flow of gases in the furnace (mixing by induction, by acceleration, or by change of direction), but above all on the nature of the atmosphere that must be maintained in the furnace. The effect of furnace atmosphere on various metals is discussed in Volume II.

In furnaces that are equipped with regenerators or recuperators, part of the potential heat in the unburned fuel that passes out of the furnace can be recovered by admission of air immediately beyond the furnace. The volume of air so admitted must be closely controlled. Too much secondary air defeats the object of heat salvage by chilling the products of combustion.

10. FUEL CONSUMPTION OF SIMPLE "IN-AND-OUT" FURNACES

The fuel economy of furnaces is commonly expressed as the quantity of fuel (or electrical energy) expended to heat a unit weight of stock, while the thermal efficiency of furnaces is expressed by the ratio:

$$\frac{\text{Heat in the stock}}{\text{Heat in the fuel consumed for heating the stock}}$$

In electric furnaces, the denominator in this ratio is replaced by the heat value of the electric energy consumed. In heating-furnace practice, such terms as "gallons of oil per ton of steel," or "pounds of coal per ton of ingots," or "kilowatt-hours per ton of material heated" are quite common.

From a study of heat losses in the preceding paragraphs, it is evident that the heat efficiency of a furnace depends not only on its design, but also, to a large extent, on its operation and on the requirements for uniformity of heating. If, for instance, a few small pieces are heated in a large furnace, the fuel consumption per unit of material heated must be extremely high, no matter whether the furnace is heated up especially for these pieces or whether it is kept hot all the time. In the former case, a large part of the heat is used to raise the temperature of the walls of the furnace; in the latter, the continued heat losses, by radiation and convection, and the continued flue-gas losses depress the heat efficiency to an extremely low value. Builders of furnaces know these facts well and are careful to make their efficiency guarantees quite specific with regard to operation. Furthermore, the human element enters very largely into the problem of heat efficiency. Furnaces are occasionally operated with partly opened or broken doors, or with leaky doors. In combustion furnaces, there may be an excess of air or excess of fuel or poor mixing. Cold air may be taken into the furnace because the stack damper is opened too wide, or unconsumed fuel may escape from the furnace because the stack damper has been lowered too much. In most modern furnaces, the effects of the human element have been minimized by automatic control of furnace pressure, furnace temperature, and fuel-to-air ratio. These controls are discussed in Volume II.

The general method of calculating the fuel consumption of a given furnace, heating a given material, includes the following steps. First, the amount of heat going to the different places, as indicated by the arrows 1 to 6, Fig. 77, must be determined for a complete heating cycle, care being taken not to omit a single item. Second, that fraction of the potential heat of the fuel that is available for heating the furnace must be found. In electrically heated furnaces, this refers to the thermal efficiency of the prime mover, the electric generator, and the transmission; in combustion-type furnaces, it refers to the difference between heating value of the fuel and the heat carried off by the products of combustion. The quotient:

$$\frac{\text{Heat required by the furnace}}{\text{Heat available per unit quantity of fuel}} = \text{Fuel requirement of the furnace}$$

may be expressed in quantity of fuel per hour, or per day, or per week, or per ton of heated material, but it should always include the whole heating cycle, including heating up and "keeping-the-furnace-waiting" time.

Occasionally the fuel consumption is split into two terms: One, called the "furnace constant" or the "holding consumption," equals the fuel needed to keep the empty furnace hot. The other equals the additional fuel consumption needed to heat the stock alone, regardless of the furnace. Any such division may be convenient for purposes of comparison, but it has no other merit.

11. THE FUEL ECONOMY OF CONTINUOUS FURNACES

In Chapter 4, the effect of firing methods on the heating capacity of continuous furnaces is discussed. The firing methods described in that chapter also exert an influence on fuel economy. Rate of heating and effectiveness of insulation of furnaces have an equally great effect.

If a continuous furnace is so designed and operated that it carries a uniform temperature of products of combustion from one end to the other, its fuel economy is equal to that of a batch-type furnace, because in that case it makes no difference whether the stock is discharged from the door through which it entered or from another door.

In the majority of continuous furnaces, the temperature varies from one end to the other. In the original continuous furnace of the pusher-type (see Fig. 2, Chapter 1), heat is released at the discharge end of the stock and the furnace temperature is lowest at the place where the stock enters. Thus, the furnace is a heat exchanger, with the hot gases flowing in one direction and the stock traveling (at intervals) in the opposite direction.

Since a frontfired continuous furnace is a counterflow heat exchanger, mathematically inclined furnace engineers have applied equations of heat exchangers to the calculation of the heat economy of frontfired continuous furnaces for heating steel or copper. The answers produced by these equations are, at best, only crude approximations, for various reasons. Heat is released by continued combustion while the gases travel through the furnace, whereas the theory of heat exchangers is based on all of the heat having been released before the gases enter. Furthermore, the theory of heat exchangers requires that the coefficient of heat transfer be constant along the furnace, whereas, as explained in Chapter 4, the coefficient is not constant in continuous furnaces, here under discussion. In spite of these discrepancies, an application of the theory is here given by an example.

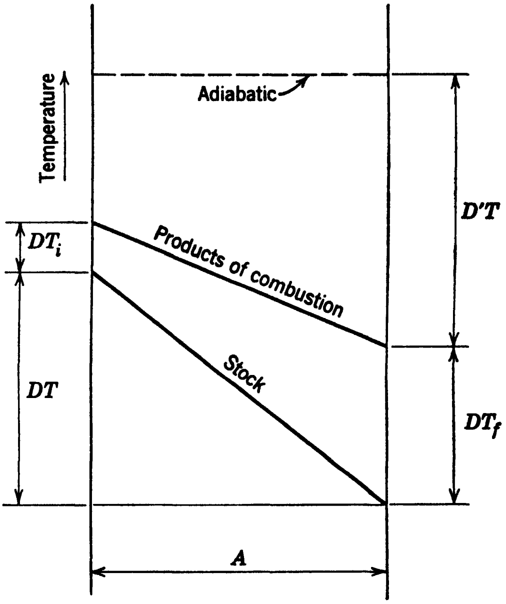


Fig. 128. Simplified diagram of heat exchange in continuous furnaces.

With the notations of Fig. 128, the heat exchange per hour is found from the equation

$$Q = kA\,\frac{DT_i - DT_f}{\ln DT_i/DT_f}$$

Q also equals the hourly heat lost by the hot substance and the heat gained per hour by the cold substance. In either case, this heat equals weight times specific heat times change of temperature.

In the heating of steel to 2200 F, let the steel enter at 100 F, and let the products of combustion leave at temperatures of either 600 F, or 1000 F, or 1400 F, or 1800 F. Furnace temperature is 2400 F. Then Table 12 shows the steps in the calculation. The values of k are estimated from experience and from values given in earlier chapters.

In the calculation, wall loss is estimated. The Btu in the fuel (lower

TABLE 12

Waste-gas temp. F	600	1,000	1,400	1,800
DT_i/DT_f	0.4	0.22	0.154	0.118
$\ln DT_i/DT_f$	−0.91	−1.56	−1.87	−2.15
$DT_i - DT_f$	−300	−700	−1,100	−1,500
$(DT_i - DT_f)/\ln \frac{DT_i}{DT_f}$	330	445	590	700
k estim. Btu/(sq ft, hr, F)	24	27	30	33
Q Btu/(sq ft, hr)	7,900	12,000	17,700	23,100
Heat loss through walls and doors Btu/(sq ft, hr)	3,200	3,300	3,400	3,500
Btu/(sq ft, hr) for heating steel $= Q'$	4,700	8,700	14,300	19,600
Lb heated/(hr, sq ft) $= Q'/(2100 \times 0.16)$	14	26	42	58
T (adiabatic) − T (final), F	3,200	2,800	2,400	2,000
$\frac{\text{Lb of prod. of comb.}}{\text{sq ft, hr}} = \frac{Q}{0.26 \times D'T}$	9.5	16.3	28.4	44.4
Btu in fuel to produce, $\frac{\text{Prod. of comb.}}{\text{sq ft, hr}}$	11,000	19,000	33,000	51,400
Btu/ton of steel	1,570,000	1,460,000	1,570,000	1,770,000

heating value) to produce one pound of products of combustion was taken to be 1160.

Although, as previously mentioned, the results of the calculation in Table 12 are only approximations, they reveal a trend which is seldom, if ever, uncovered in practice, because the owner of a furnace tries to heat as much metal as is compatible with good heating, and, therefore, is not interested in the change of fuel economy at lower rates of heating.

The method of calculation shown in Table 12 furnishes a very close approximation in the heating of aluminum and similar metals, because the coefficient of heat transfer is more nearly constant and because no combustion occurs in the furnace.

A more accurate calculation of the fuel consumption of a frontfired continuous furnace for heating steel takes much more time than the calculation of Table 12 and is based on assumptions that must be so selected that they come close to actual conditions. The calculation is long, because the furnace length must be divided into sections within each of which the coefficient of heat transfer and the wall loss are practically constant. The assumptions deal not only with temperature and emissivity of the flame, but also with heat losses through walls and hearth and to water-cooled parts, if such are installed.

The method will be explained by applying it to the simplest type of continuous furnace, namely, an endfired furnace with heat above the steel only. The steel is to be so thin that temperature difference between top and bottom need not be considered. This means that the thickness must not exceed 4 in.

A practical furnace length is selected, based on heating rates given in Chapter 4. A number of temperatures are selected for the products of combustion at the point of leaving the furnace (and for the heating stock). For each of these "leaving" temperatures, a temperature distribution throughout the length of the furnace is tentatively sketched for the products of combustion and for the stock, on the basis of experience gained from previous calculations or test data. The necessity for assuming tentative temperature distributions is due to the fact that the ratio of wall losses to useful heat varies with the temperature distribution and with the design of the wall. With these assumed temperature distributions, the transfer of heat is calculated by making use of the laws of heat transmission, and a new temperature distribution of steel and of products of combustion is determined by calculation. If that distribution deviates much from the original assumption, the calculation must be repeated. If the coincidence of assumed and of calculated distributions is good, the rate of heating and the fuel consumption per pound of steel obtained from the first calculation are correct.

As previously stated, several assumptions compatible with practical conditions are necessary. In this example, it is assumed that the furnace is constructed of dense firebrick, with a roof 9 in. thick, with sidewalls 13½ in. thick, and with a ventilated hearth 18 in. thick. It is also assumed that the coefficient of heat transfer by convection equals 2 Btu per (sq ft, hr, F).

Selection of the coefficient of heat transfer (in each section) requires careful consideration. In the early sections, where combustion is still going on, the coefficient may slightly exceed the values of Table 5. In other sections, it drops below those values, because of low emissivity of the roof and of the stock.

Since the calculation is very long, it is here carried out for one terminal gas temperature only, namely, 1800 F. The steel is heated from 60 F to 2200 F. The fuel is raw producer gas, entering at 1100 F. It is burned without excess air. The waste gases contain 2 per cent

CO plus 2 per cent H_2. The calculations are based on 1 lb of steel, or on 1 sq ft, and are, therefore, applicable to all sizes of furnaces of similar shape.

From Fig. 116, lower calorific value of raw producer gas = 137.3 Btu per cu ft. Sensible heat in gas at 1100 F = 21.0 Btu. Sum = 158.3 Btu per cu ft. By extrapolating curve *A* of Fig. 116, the adiabatic flame temperature (i.e., the temperature at which the sensible heat content of the products = 158.3) is found to be 3100 F. The ratio of actual to ideal temperature rise during combustion is found in practice to be about 77 per cent. The equivalent temperature of gas plus air, if mixed before combustion, would be 550 F (from Fig. 116). Hence, actual maximum flame temperature = 550 + 77% × (3100 − 550) = 2510 F.

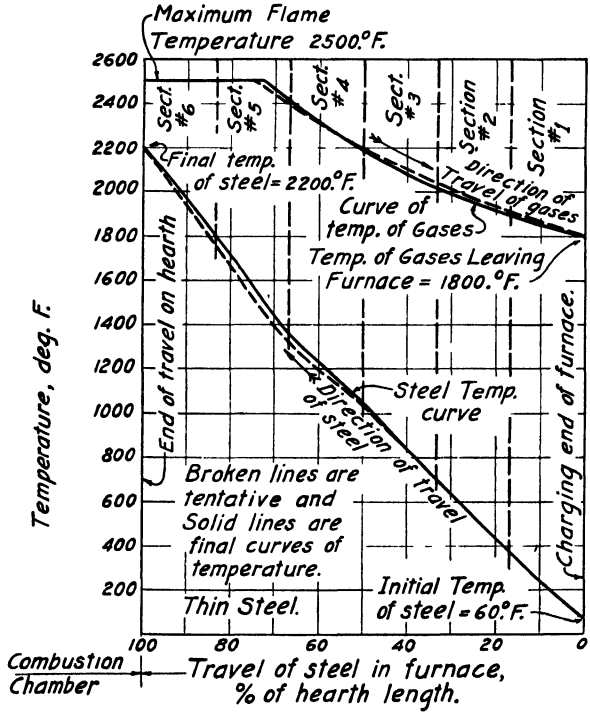


Fig. 129. Illustration of method of calculating temperature distribution in a continuous front-fired furnace.

It is necessary, first, to assume tentative temperature-distribution curves, as in Fig. 129, relating temperature of steel and temperature of flame to position in the length of the furnace, for the purpose of determining the radiation losses. If the latter were a constant fraction of the heat input to the steel, this procedure would not be necessary; actually, however, the ratio of heat lost through the walls to heat absorbed by the steel is not the same in different zones of the furnace. It also varies with the rate of heating. The tentative temperature-distribution curves* are shown in broken lines in Fig. 129. The length of the hearth (which is called 100 per cent and therefore need not be known in feet) is first divided into a number of equal lengths, and the ratio of wall losses to heat input to the steel in each section is figured as in Table 13, based on the assumed temperatures.

TABLE 13

1 Sect. No.	2 Initial and Final Temperature of Steel in Section (from Tentative Curve), F	3 Heat Input per lb of Steel in Section (Fig. 27), Btu	4 Average Temperature of Flame and of Steel in Section (Tentative), F	5 Average Wall Temperature in Section, F	6 Wall Loss in Section per sq ft of Steel Surface, Btu per hr	7 Quantity of Heat Put into Steel in Section, per sq ft of Steel Surface, Btu per hr	8 Ratio of Wall Losses to Heat Input to Steel, %
1	60 370	32	1850 215	1030	930	2,880	32.3
2	370 690	42	1965 530	1300	1,330	3,780	35.1
3	690 1030	52	2110 860	1590	1,850	4,680	39.5
4	1030 1310	64	2290 1170	1870	2,420	5,760	42.0
5	1310 1750	92	2460 1530	2140	3,400	8,280	41.1
6	1750 2200	81	2500 1975	2350	4,280	7,290	58.8
					Sum = 14,210	Sum = 32,670	

The greater the number of sections into which the furnace length is divided, the more accurate is the calculation, but the use of six sections keeps the calculation within reasonable limits and, at the same time, gives sufficient accuracy for all practical purposes.

The calculation is started from the cold end rather than the hot end of the furnace, because it is necessary to work back from the final temperature of the gases (which was assumed and is, therefore, a given quantity in the calculation) to the temperatures at other points, which are to be determined.

$$\text{Average ratio } \frac{\text{Wall losses}}{\text{Heat input to steel}} \text{ for whole furnace} = \frac{14{,}210}{32{,}670} = 43.3\%$$

The values for column 3 in this table are read from Fig. 27. The average steel temperature is taken as the arithmetical mean between initial and final temperatures for the section. In column 5 the wall temperature is intermediate between flame temperature and steel temperature; it is taken equal to [steel temperature + *f* × (flame temperature − steel temperature)], and, from previous calculations, the values of *f* are found to be approximately as follows:

---

* Actually, the temperature curve is not flat on top but reaches a maximum and then decreases. For the present purpose, however, a constant maximum temperature can be assumed without appreciable error.

Section No.	1	2	3	4	5	6
f	0.50	0.54	0.58	0.62	0.66	0.72

For column 6, the heat loss for the 9-in. arch and 13½-in. sidewalls is read from Fig. 86, for temperatures as in column 5, and the loss from the hearth is taken as 70 per cent of the values for an 18-in. wall, but at the temperature of the steel. Furthermore, it is assumed that the steel covers only 85 per cent of the hearth width, and that the height of the sidewalls (two sides) as a fraction of the hearth width is as follows:

Section No.	1	2	3	4	5	6
Height of 2 sidewalls as fraction of hearth width	0.40	0.50	0.60	0.70	1.00	1.00

A rough calculation, based on the tentative temperature difference of Fig. 129, shows that the heating rate to be expected is about 90 lb per (sq ft, hr). The values of column 7, then, are those of column 3 × 90. The values of column 6 divided by those of column 7 give the ratios of wall losses to useful heat input for the various sections as in column 8.

The next step is the determination of the quantity of gases passing through the furnace. From Fig. 116, the lower calorific value of raw producer gas is 137.3 Btu per cu ft of dry gas (referred to 60 F). The incomplete combustion loss, with 2 per cent CO + 2 per cent H_2 and no excess air, is 20 Btu (Fig. 116). The sensible heat in the fuel gas at 1100 F is 21 Btu (Fig. 116, curve *C*). If the products of combustion leave at 1800 F, the sensible heat that they carry out of the furnace amounts to 79 Btu (Fig. 116). Hence the available heat is $(137.3 + 21.0 - 79.0 - 20.0) = 59.3$ Btu per cu ft of producer gas. The heat input to the steel, from 60 to 2200 F = 363 Btu, and from the preceding tabulation the radiation loss is 43.3 per cent. Hence $(363 \times 143.3\%)/59.3 = 8.76$ cu ft of producer gas is required per lb of steel heated. From Fig. 116 the lower heating value of raw producer gas is read to be 137.3 Btu/cu ft. Hence, the heat consumption is $8.76 \times 137.3 \times 2000 = 2{,}400{,}000$ Btu per ton. From curve *A* of Fig. 116, the (average) sensible heat content of the products = 0.0525 Btu per F for each cubic foot of producer gas; hence $8.76 \times 0.0525 = 0.460$ Btu per lb steel = heat given up by the products of combustion in cooling 1 F.

The next step is the determination of the relation of flame temperatures to steel temperatures, and of both to position in the furnace. In Table 14 (in which the sections are not necessarily of equal length), column 10 is the same as column 3 above, and represents useful heat per pound of steel.

The sum of useful heat plus radiation losses (column 11), obtained by multiplying the values of column 3 by (100 per cent plus ratio for column 8, in percentage), must equal the heat given up by the gases. Hence dividing the values of column 11 by 0.460 Btu per F gives the temperature drop of

TABLE 14 (Continuation of Table 13)

9	10	11	12	13	14	15	16	17	18	19
Sect. No.*	Heat Input Btu per lb of Steel, in Section (Fig. 27)	Heat to Steel + Wall Loss in Section, Btu per lb	Temperature Drop of Gases in Section, F	Initial Temperature† of Gases in Section, F	Average Flame Temperature and Average Steel Temperature, F	Equivalent Emissivity of Flame+Roof	Rate of Heat Transfer to Steel, Btu per (sq ft, hr)	Time Required in Section, hr per psf	Length of Section, % of Hearth Length	Distance from Cold end, % of Hearth Length
1	32	42.5	92	1892	1846 215	0.37	17,900	0.00179	16.2	16.2
2	42	56.7	123	2015	1954 530	0.39	22,200	0.00189	17.1	33.3
3	52	72.3	157	2172	2094 860	0.43	29,300	0.00177	16.0	49.3
4	64	90.9	197	2369	2271 1170	0.47	39,300	0.00163	14.8	64.1
5	92	129.6	(281)	2500	2440 1530	0.51	48,200	0.00191	17.3	81.4
6	81	128.8	(280)	2500	2500 1975	0.55	39,500	0.00205 Sum = 0.01104	18.6 Sum = 100.0%	100.0

Heating rate = 1/0.01104 = 90.5 lb per (sq, ft, hr)

* In this table, the sections are not all of equal length as in Table 13. Instead, each section corresponds to the (unknown) length in which the temperature rise of the steel marked in column 2, Table 13 is effected, and the calculation of this length (as in column 18) is one of the purposes of this tabulation.

† "Initial temperature" in this column means initial in reference to the direction of gas travel; or it is the highest temperature of gases in the section.

the gases in each section (column 12), and from these the initial, final, and average temperatures of the combustion gases are figured (columns 13 and 14). The average steel temperature is the same as in column 4.

In figuring the heat transfer, the emissivity of the luminous flame of producer gas has been assigned the values given below, corresponding to test results with similar flames, and the total equivalent emissivity of flame plus walls (including heat transfer by convection) was calculated on the basis that the flame absorbs part of the roof radiation.

	1 (cold end)	2	3	4	5	6 (hot end)
Emissivity of flame only, % of blackbody	19	21	23	25	27	30
Total equivalent emissivity of flame + roof E, %	37	39	43	47	51	55

The rate of heat transfer (column 16), in Btu per (sq ft, hr), is

$$0.172 \times E\left[\left(\frac{\text{avg. flame temp., F} + 460}{100}\right)^4 - \left(\frac{\text{avg. steel temp., F} + 460}{100}\right)^4\right]$$

Dividing the heat input to the steel (column 10) by the heat-transfer rate (column 16) gives the time required per pound of steel per square foot of exposed area (column 17). Dividing the separate values of this column by their total gives the fraction of the total time required for each interval of temperature rise (column 18), and, because the rate of travel of the steel through the furnace is the same throughout the length, the values of column 18 also represent the fraction of the hearth length actually corresponding to each section. By plotting the steel temperatures from column 2 and the flame temperatures from column 13 against position in the furnace (column 18), the solid-line curves of Fig. 129 are obtained.

The sum of the heating times for all sections (column 17) is 0.01104 hr per (psf); hence the rate of heating is 1 ÷ 0.01104 = 90.5 lb per (sq ft, hr), based on the exposed area of the steel. Since this is so nearly the same as the initially assumed value of 90 lb per (sq ft, hr) and the final temperature curves it is unnecessary to repeat the calculation.

Because the calculated temperature distribution almost coincides with the assumed temperatures, the above calculated gas consumption per pound of steel and the heat consumption per ton of steel are correct.

If the same method of calculation is pursued for other exit temperatures of the products of combustion, and if the results are plotted, Fig. 130 is obtained. The curves differ considerably from the logarithmic curves of temperatures that are found if the coefficient of heat transfer is constant.

Obviously the calculation is quite long, even for a single exit temperature of the waste gases. It would be much longer, if other modifying factors had been included.

A similar set of calculations for the same furnace, this time fired with coke oven gas, furnished results that were plotted in Fig. 70.

The principal difference from the earlier calculation consists in figuring with radiation from clear gases.

At first thought, it might be judged that a furnace of a given size, with both top and bottom heat can heat twice as much steel as a top-fired furnace and do it with the same fuel consumption per unit weight of steel heated. Actually, matters are different. It is difficult to pass as much weight of products of combustion underneath as can flow above, unless the stock rests on water-cooled skid pipes which are supported by tall water-cooled stanchions. These water-cooled parts absorb much heat not only from the flame gases, but also from the steel whereby fuel consumption per pound of heated material is in-

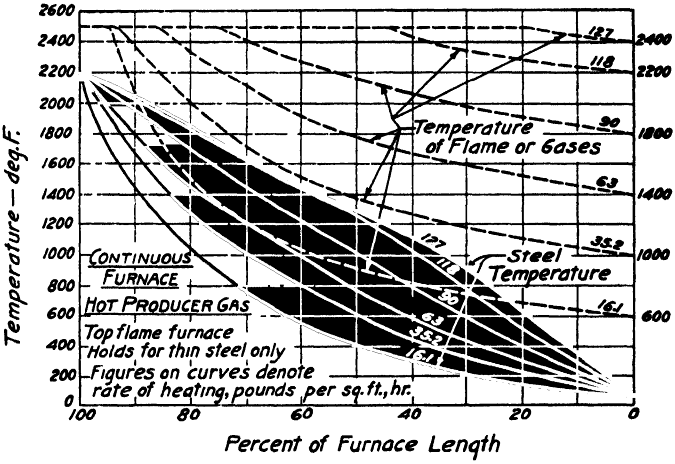


Fig. 130. Temperature variation of steel and of gases in a continuous furnace heated by producer gas.

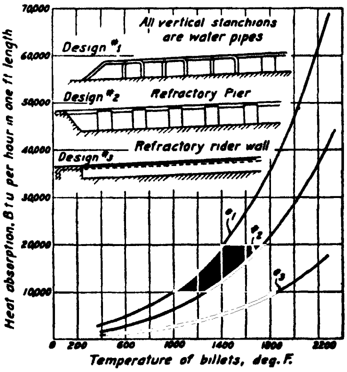


Fig. 131. Heat absorbed by water-cooled skid rails. W. Heiligenstaedt, *Archiv für das Eisenhüttenwesen,* May 1932, page 559.

creased. Figure 131 contains information on heat absorbed by water-cooled skid rails and stanchions. Compare also Chapter 4. A calculation, similar to the one that led to the plotting of Fig. 130, can be made for an endfired furnace with top and bottom heat. However, such a calculation is extremely long, and has no value, because such furnaces are not built without a soaking zone.

12. EFFECT OF STOCK THICKNESS ON THE FUEL ECONOMY OF CONTINUOUS FURNACES

If material with low emissivity and high thermal conductivity is heated, thickness of stock does not affect fuel economy, because the heat absorbed by the surface of the stock is transmitted to the interior with but a negligible temperature drop. Heating of aluminum is an example. Contrariwise, if the emissivity of the stock is high and its thermal conductivity is low, as in steel, thickness affects fuel economy, for the following reason: Since the stock is to be heated to a given average temperature, the surface is hotter than the interior and the products of combustion, for the same rate of heating to the same average temperature, must leave with a higher temperature than they do when heating thin stock. Conversely, if the gases are to leave at the same temperature regardless of thickness of stock, the thick material must stay in the furnace a longer time than thin stock. (Compare Fig. 68.) Expressed in other words: The rate of heating must be reduced, whereby the wall loss per unit of weight heated is increased. If the heating stock is easily oxidized, other factors enter. Scale has a higher emissivity than bright metal. In the initial stages of heating, it promotes absorption of heat; but thick scale formed during prolonged heating of thick stock is an insulator which, in turn, requires the stock to stay in the furnace a longer time. And if the heater attempts to increase the rate of heating by increasing the supply of heat, the scale is softened, becomes shiny and reflects the heat which means that its emissivity is lowered.

Rate of heating and fuel economy in the heating of thick steel can be calculated in the manner explained above, if an additional estimate is made—namely, the difference between surface temperature and average temperature of the stock. Care and judgment are needed in arriving at a fairly correct estimate, because of variations in the conductivity of steel as a function of composition and temperature. The results of such a calculation for heating mild steel of 6-in. thickness are given in Fig. 132. The calculation had to be made twice, because the estimated temperatures differed from the calculated values.

Since a theoretical calculation starts with an estimate that is based on experience, furnace engineers prefer to collect data on fuel economy of furnaces of different types. The engineers base estimates, or even guarantees, for a new furnace on data obtained from existing similar furnaces operating under similar conditions. Theory is then applied to the data to obtain a fairly correct answer on a projected furnace. An often heard opinion is that a good guess, based on experience, is to be preferred to a long theoretical calculation.

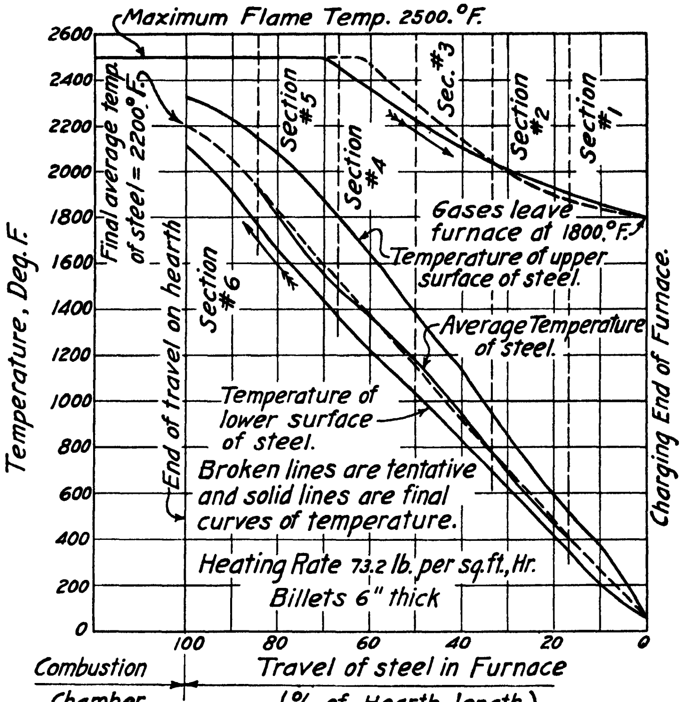


Fig. 132. Calculated temperature distribution in a continuous furnace heating steel billets 6 in. thick.

Calculations on fuel economy are even longer for triple-fired furnaces, quintuple-fired furnaces (erroneously called five-zone furnaces), and for furnaces with partial side firing. The same statement is true for doughnut furnaces, such as illustrated by Fig. 133, especially if such furnaces are subdivided into zones by baffles. The zones permit control of rate of heating at various levels of temperature.

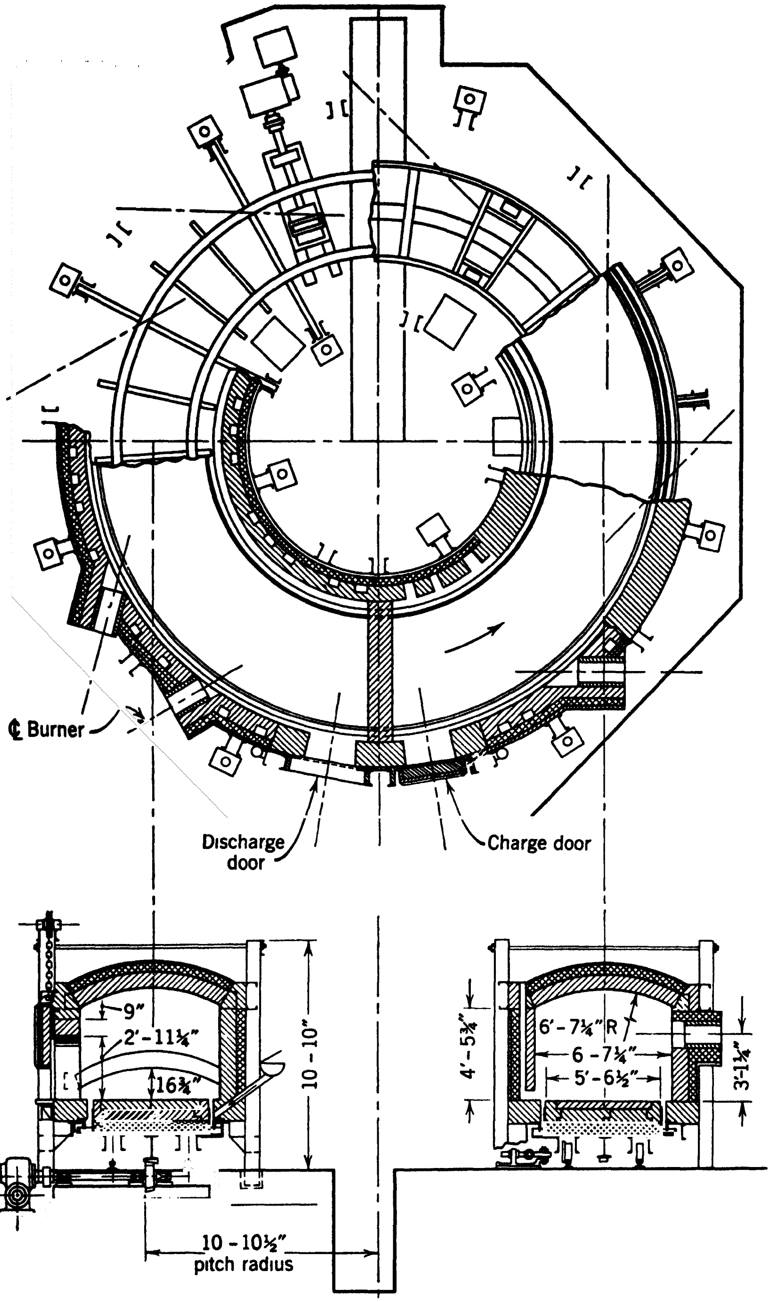


Fig. 133. Large furnace with rotating hearth ("doughnut" type) with burners on outer circumference.

TABLE 15

Input of heat	Btu per hr	%
Heat of combustion of tar (gross calorific value)	48,170,000	
Heat of combustion of coke-oven gas (gross calorific value)	19,240,000	
Heat in steam used for atomizing tar	2,745,000	
Total	70,155,000	100%
Output of heat		
Heat imparted to steel slabs	35,200,000	50.2
Heat lost by conduction through walls ("radiation loss")	4,170,000	5.9
Heat absorbed by water-cooled skid rails and supports	6,310,000	9.0
Sensible heat lost in stack gases	19,040,000	27.2
Latent heat of water vapor produced by combustion, and of steam used for atomizing tar	5,833,000	8.3
Unaccounted for	−395,000	−0.6
Total	70,158,000	100.0%

Thermal efficiency = 35.2/70.16 = 50.2%

Other data from the test are the following:

Thickness of slabs	5 in.
Weight of steel heated	94,000 lb/hr
Tar burned in heating zones:	
Upper burners 221⅛ gal per hr	
Lower burners 101 gal per hr	322⅛ gal per hr
Gas burned in soaking zones (referred to 60 F, 29.92" Hg abs)	35.640 cu ft per hr
Calorific value of tar: Gross	149,400 Btu per gal
Net	144,200 Btu per gal
Calorific value of gas: Gross	540 Btu per cu ft
Net	482 Btu per cu ft
Total combustion heat of fuel (gross calorific value)	67,400,000 Btu per hr
Steam used for atomizing	1980 lb per hr
Steam pressure at burners, average	75 psi
Water used for cooling	360,700 lb per hr
Temperature rise of water	17.5 F
Weight of gases flowing in stack	100,000 lb per hr
Ideal quantity of air required for combustion	49,330 lb per hr
Excess air in stack	95%
Weight of air supplied to furnace by recuperator	51,600 lb per hr
Excess air supplied to furnace	4.6%
Temperature of gases at base of stack	770 F
Temperature of air entering recuperator	74 F
Temperature of air leaving recuperator	525 F
Temperature of gases entering recuperator	1212 F
Temperature of gases between first and second passes of recuperator	918 F
Temperature of gases leaving recuperator	809 F
Average temperature of walls in soaking chamber of furnace	2320 F
Highest temperature of walls in upper heating chamber of furnace	2410 F
Temperature at door 8 in furnace	2150 F
Temperature of billets leaving furnace	2250 F
Temperature of billets entering furnace	50 F
Temperature at cold end of furnace	1233 F

Rate of heating, psf of hearth, hr:	
Including soaking hearth	66.22
Not including soaking hearth	87.0
Including top and bottom surface and soaking hearth	37.2

Gas analysis:	% CO_2	% CO
Heating zone, door 6	15.4	0.0
Heating zone, door 8	11.0	0.1
Heating zone, cold end	14.8	0.0
Base of stack	7.7	
Heat recovered by recuperator	8.3% of heat of fuel	

Test data from actual furnaces give a good idea of the distribution of heat and also of fuel economy. Table 15 is the record of a test made on a triple-fired slab-heating furnace, such as illustrated in Figs. 69 and 318.

Several items are worthy of discussion. The rate of heating is rather low, if it is based on the whole heat-receiving area (top, bottom, and soaking chamber). The thermal efficiency (ratio of heat in steel to heat in fuel) is 50 per cent, which is low for so elaborate a furnace. Heat recovery by the recuperator is also low, namely, 8.3 per cent. The low efficiency is, in part, due to an explosion that had occurred in the recuperator; the latter had been repaired, but not entirely, and had some leaky spots.

In the years that have elapsed since the test was made, the heating capacity of the furnace was, at different times, increased by changes in the burners, the width of the hearth, and in the stack.

Furnaces of this general type are capable of a much higher thermal efficiency, which fact is proved by a publication in *Stahl und Eisen,* January 19, 1939; it concerns a furnace similar to the one shown in Fig. 318. The furnace was erected in Dinslaken, Germany. With heating rates referred to the sum of exposed surfaces in top- and bottom-heating chambers, the following efficiencies were observed in day-to-day operation:

Heating rate, lb per (sq ft, hr)	20.45	40.9	51.1
Thermal efficiency %, heat in steel/heat in fuel	60.2	70.5	73.4

Combustion air for the main heating chambers was preheated to 1040 F in a metallic, all-welded and, therefore, airtight recuperator. The recuperator caused a saving of heat of more than 16 per cent. Air for the soaking chamber was not preheated. Water in the skid pipes was condensate (distilled water), which can be heated to almost 170 F. Under the skid pipes (over the underfiring chamber) arches were built which keep most of the radiation away from the water-cooled pipes. In consequence, heat loss to the skid pipes and their supports was only 2.7 per cent of the heat in the fuel, which is a very low value. The arches must be rebuilt every 4½ to 5 months. American furnace owners may not wish to put up with frequent shutdowns. Gas and air were delivered in always constant ratio by displacement pumps which were geared together.

Heiligenstaedt (*loc. cit.*) made a detailed analysis of the performance of this furnace and arrived at the following percentages.

Item of Heat	Per Cent
Heat to steel	74.7
Wall loss	3.7
Heat to skid pipes	2.7
Wall loss of recuperator	1.0
Wall loss of hot-air duct	0.6
Leakage loss (charging door, etc.)	7.9
Stack loss	9.4
	100.0

13. FUEL ECONOMY OF SIDEFIRED CONTINUOUS FURNACES

Sidefired continuous furnaces, such as illustrated in Fig. 134, have been used for many years in forge shops and in the practice of heat-treating. In rolling mills in the United States, side firing of continuous furnace was somewhat hesitatingly introduced after 1950. Some engineers were afraid of temperature differences across the width of the furnace. Others claimed that the side burners would interfere with the trouble doors. Both apprehensions were found to be groundless.

In addition to the usual factors, such as rate of heating, final temperature of stock, type and thickness of refractories, other factors affect the fuel economy. Such factors are: Distribution of heat input along the furnace and location of offtake or vent.

Evidently, side firing is practically equivalent to end firing, if all the burners at the hot end are wide-open, while the burners at the cold end are either closed or turned to low fire. However, side-fired furnaces are seldom operated in that manner. Whenever the composition of the heating stock permits, the burners at the cold end fire wide-open, to impart as much heat as possible to the entering stock, thereby increasing furnace capacity with good uniformity of temperature in thick stock. It is clear that, with heavy firing at the entering end, the products of combustion leave the furnace with a higher temperature than they do in an end-fired furnace, and that a higher fuel consumption is the price that is paid for increased heating capacity coupled with good uniformity of temperature (see Fig. 66). The effect of a high exit temperature of the burned fuel gases is most marked with fuels that burn with a low adiabatic flame temperature—for instance, blast furnace gas and producer gas. For that reason, the sidefired furnace is here compared with an endfired furnace having

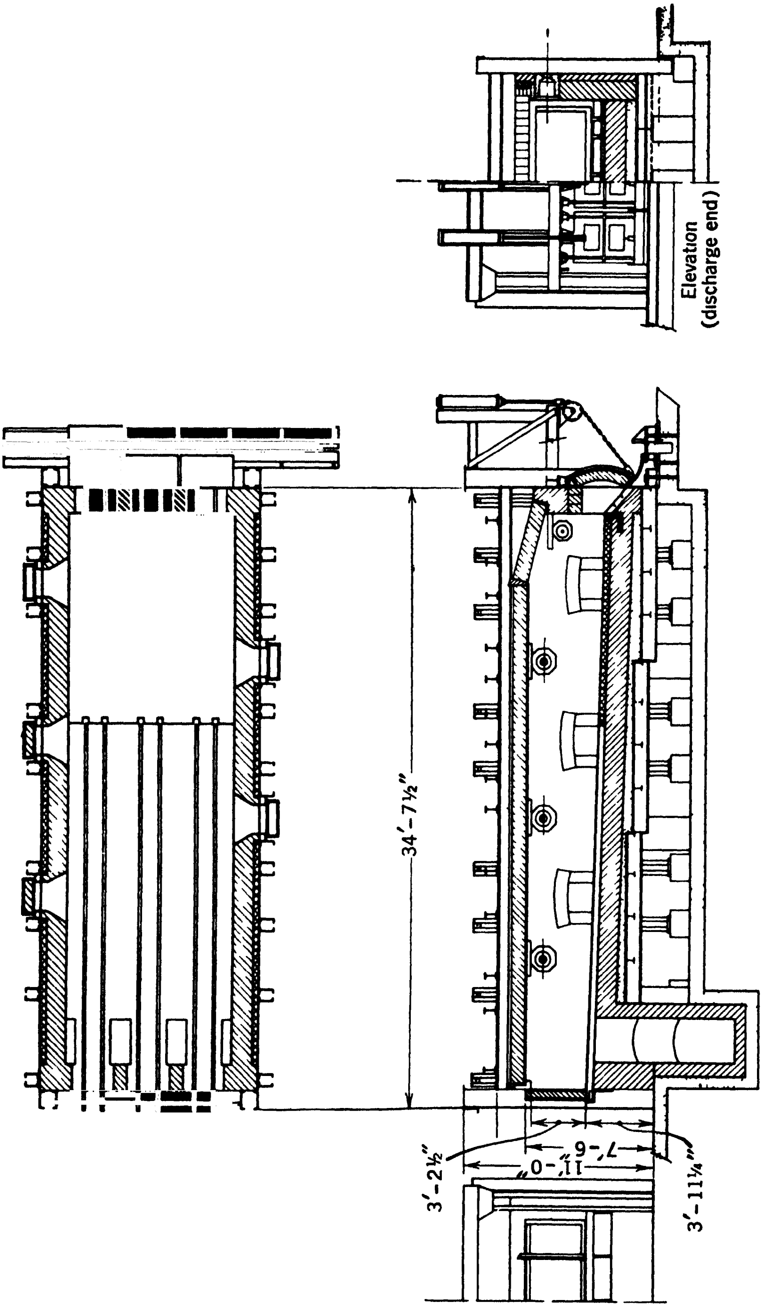


Fig. 134. Sidefired continuous furnace

the temperature distribution of Fig. 132. The high wall loss of that furnace makes the difference in fuel economy quite conspicuous.

For the heating rate indicated in Fig. 132, the surface temperature of the stock at the hot end is about 60 F higher than the average temperature (see Fig. 68). The ratio of stock surface to surface of refractories is approximately 0.4. From Fig. 52, the coefficient of heat transfer for 2200 + 60 F = 26.7 Btu per (sq ft, hr, F). But, since the flame of raw producer gas is brightly luminous, the coefficient k must be increased at least 10 per cent. This increase raises k to 29.4. For a trial, let the average temperature of the flame equal 2450 F, then the active temperature head equals $2450 - 2260 + \frac{1}{3}(2260 - 60) = 923$ F. Heat transferred per (sq ft, hr) = 29.4×923 Btu. Heat to be imparted per lb = 363 Btu. Then heating time follows from $t \times 29.4 \times 923 = 245 \times 363$; $t = 3.27$ hr. Steel surface is referred to in 245/3.27 = 75 lb per (sq ft, hr). This is close enough to the rate of Fig. 132.

Since producer gas that is burned with cold air has a low flame temperature, the fuel economy of a sidefired furnace burning producer gas varies with the direction of the flow of the products of combustion. If these products leave at the hot end, their temperature is 2450 F or higher. If they leave at the cold end, their temperature cannot exceed 2300, because the cold charge absorbs heat rapidly. The heat left in the furnace per cubic foot of gas equals combustion heat plus sensible heat in fuel gas minus (sensible heat carried out by leaving products of combustion plus heat lost by incomplete combustion). If, as in former examples, the lower heating value equals 137.3 Btu per cu ft, the sensible heat in the gas equals 21 Btu per cu ft, and the not released heat (incomplete combustion) equals 20 Btu per cu ft, then the heat left in the furnace per cubic foot of fuel gas equals 30.3 Btu per cu ft of fuel gas, if the products of combustion leave the furnace at 2300 F. If they leave at 2450 F, only 18.3 Btu are left in the furnace per cubic foot of fuel gas. As in earlier examples, 363 Btu are needed to heat a pound of steel and since the ratio of wall loss to useful heat is approximately 45 per cent, the heat left in the furnace per pound of steel equals 1.45×363 Btu. If the products of combustion leave the furnace at the cold end, the heat consumption per ton of steel equals

$$\frac{1.45 \times 363 \times 2000}{30.3} = 3.5 \text{ million Btu}$$

If products of combustion leave at the hot end, the heat consumption rises to 5.8 million Btu per ton.

In this example, the sidefired continuous furnace consumes much more fuel than the endfired furnace that has the temperature distribution of Fig. 132. With that temperature distribution the heat consumption per ton of steel equals 2.7 million Btu. The example is instructive. The burners were assumed to mix so poorly that much heat is wasted by incomplete combustion. Furthermore, the fuel (raw producer gas) burns with a low flame temperature in cold air: In practice, producer gas is, with few exceptions, burned in preheated air, whereby the adiabatic flame temperature is raised. The difference in fuel consumption is very small with fuels such as oil or natural gas. And finally, a furnace built up of dense firebrick without insulation is an abnormality.

The conclusion is that little difference exists between the fuel economy of endfired continuous furnaces and of sidefired furnaces, if the latter are properly designed and operated and if fuel with a high adiabatic flame temperature is burned. This statement refers to furnaces in which a temperature of about 2200 F is maintained. For furnaces with lower temperatures no noticeable difference in the fuel consumption exists for either type of firing.

The same statements are true for furnaces with rotating hearths, such as illustrated in Fig. 133. It is evident that furnaces of this type cannot be endfired, but the firing can be so arranged that it is equivalent to end firing. For metallurgical reasons, some furnaces of this type are divided into sections by radial baffles.

It may be repeated that in the United States continuous furnaces are built for labor economy. If fuel economy is an object, it is attained by recuperation.

14. EXAMPLES FOR THE CALCULATION OF FUEL CONSUMPTION

EXAMPLE 1. GASFIRED FORGE FURNACE, REGENERATIVE TYPE (SEE Fig. 135). In this furnace, ingots of medium-carbon steel and of alloy steel, 26 in. in diameter, are heated for forging. After each heating period, in order to avoid formation of cracks in the ingot by too rapid initial heating, the furnace interior is cooled down to 1200 F, before another ingot is charged. The temperature of the inside surface of the walls, then, varies according to the following cycle: Heat from 1200 to 2400 F, in 6 hr at approximately linear rate; drop from 2400 to 2300 F during soaking period of 1 hr; then cool, with door open, to 1200 F, when another ingot is charged.

The first step is to determine the temperature distribution throughout the walls for each of the various thicknesses, at the end of the cooling period and at the end of the (heating plus soaking) period. Schmidt's graphical method is to be used (see Appendix, page 444). In Fig. 136 are shown the temperature curves determined by this method for the 18-in. sidewalls. Approximately 1½ hr is required for cooling from 2300 to 1200 F, and the time average of the inside surface temperature is 1860 F. This means that, after many cycles, the loss by conduction to the outside is equal to that of an 18-in. wall

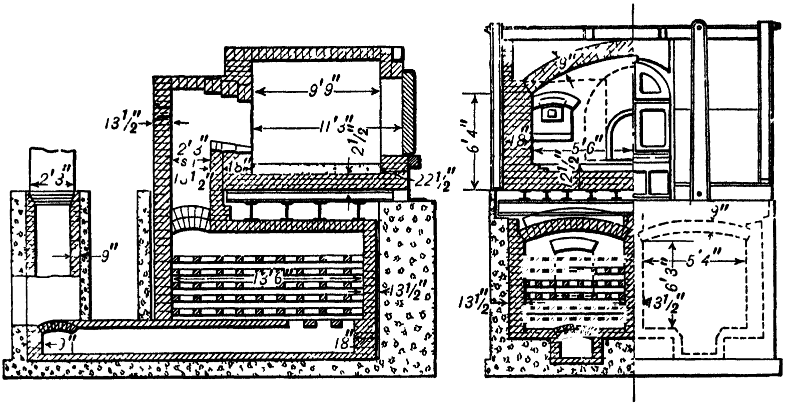


Fig. 135. Gasfired forge furnace. Regenerative type.

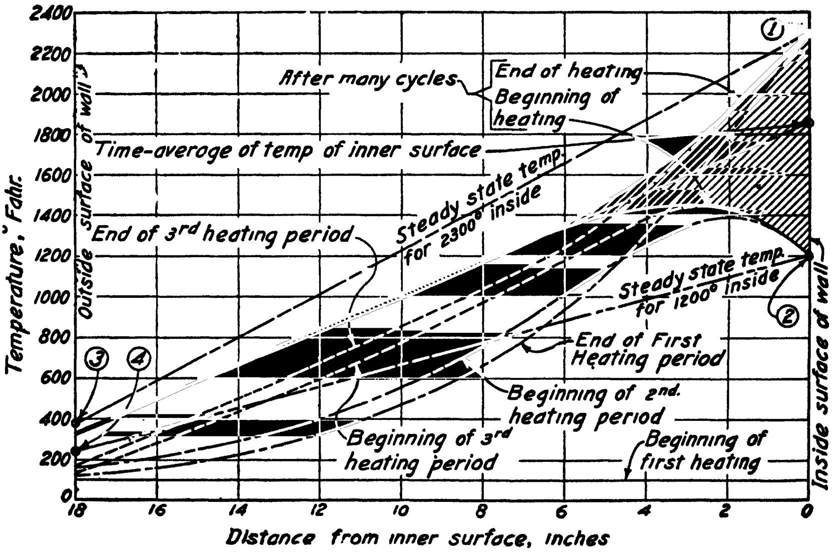


Fig. 136. Temperature variation in 18-in. firebrick wall of forge furnace during 8½-hr heating-and-cooling cycle.

with 1860 F inside temperature, or (from Fig. 86) 690 Btu per (sq ft, hr). The storage loss factor is the ratio of the heat actually lost by storage during the cycle, to that which would be lost if steady-state temperatures were reached throughout the wall at the beginning and at the end of the heating period; or, referring to Fig. 136, it is the ratio of the algebraic sum of the section-lined areas to the area of the trapezoid 1–3–4–2. For the 18-in. wall, this ratio is 21 per cent.

During the first few cycles after starting up from cold, the heat-storage loss is considerably greater; thus, for the first cycle, the storage loss factor is 101 per cent; for the second immediately following cycle, 38½ per cent; and the third immediately following cycle 29 per cent, for the 18-in. wall. On the other hand, the conduction loss moves in the opposite direction; for the first, second, and third heating periods, it is 74, 301, and 466 Btu per (sq ft, hr) respectively. The examples that follow are based on a cycle sufficiently far removed from a cold start.

For the 9-in. arch, the storage ratio after many cycles is 39 per cent; for the 12½-in. hearth it is 28 per cent; and for the 13½-in. downtake wall, 27 per cent. In the following calculations, these factors are enclosed in braces, thus {27 per cent}.

Roof Arch. 9 ft 9 in. by 11 ft 3 in. by 9 in. thick. When the furnace temperature is 2300 F (end of heating plus soaking period), the outside wall temperature is shown by Fig. 85 to be 510 F, and the average wall temperature at this time is

$$\frac{2300 + 510}{2} = 1405\text{ F}$$

Similarly, when the interior is at its lowest temperature of 1200 F, the outside temperature is 328 F and the average is 764 F.

In the following, the heat-content figures are read off from Fig. 90 and are enclosed in square brackets, thus: [354 − 172]. The heat change in the brickwork = *weight* × (*heat content per pound, average at end of heating period* − *heat content per pound, average at end of cooling period*). If equilibrium temperatures were reached throughout the thickness, the change in heat content of the roof arch would be

$$9.75\text{ ft} \times 11.25\text{ ft} \times 0.75\text{ ft} \times 125\text{ pcf} \times [354 - 172] = 1{,}872{,}000\text{ Btu}$$

Since the storage factor for this particular cycle, as stated above, is 39 per cent for the 9-in. arch, the actual heat change is

$$\{39\%\} \times 1{,}872{,}000 = 730{,}000\text{ Btu}$$

Sidewalls. Two walls, each 9 ft 9 in. by 5 ft 3 in. by 18 in. thick. From Fig. 85, temperature of outside corresponding to 2300 F inside = 377 F, and average temperature for steady-state conditions at end of (heating plus soaking) period would be 1338 F. From Fig. 90, heat content at this temperture = 332 Btu per lb. From Fig. 85, outside temperature (corresponding to 1200 F inside) = 246 F, and average temperature for steady-state conditions would be 723 F. From Fig. 90, heat content at this temperature = 162 Btu per lb.

Actual change of heat content

$$= \{21\%\} \times (2 \times 5.25 \times 9.75 \times 1.5) \times 125 \times [332 - 162] = 685{,}000\text{ Btu}$$

Ends. Two ends, each 11 ft 0 in. by 6 ft 0 in. (average height) by 18 in. thick. Door in one end, 5 ft by 5 ft by 9 in. thick. Two flues in one end, 2 ft by 2 ft = 8 sq ft.

Front end: (11 ft 0 in. × 6 ft 0 in.) − 25 = 41 sq ft

Change of heat content

$$= \{21\%\} \times 41 \times 1.5 \times 125 \times [332 - 162] = 274{,}300 \text{ Btu}$$

Door: 5 ft 0 in. by 5 ft 0 in. by 9 in. thick.

Change of heat content

$$= \{39\%\} \times 5 \times 5 \times 0.75 \times 125 \times [354 - 172] = 166{,}400 \text{ Btu}$$

Back end: (11 ft 0 in. × 6 ft 0 in.) − 8 = 58 sq ft

Change of heat content $= \{21\%\} \times 58 \times 1.5 \times 125 \times [332 - 162] = 388 \text{ Btu}$

Total heat in ends = 828,700 Btu.

Bottom. 9 ft 9 in. by 11 ft 0 in. by 12½ in. thick.

The hearth is well ventilated. Nevertheless, both radiation and convection cooling are restricted. Assuming that each is reduced to 50 per cent of the values of Fig. 79, a calculation by the methods of pages 120-121 showed that for 2300 F inside, the steady-state temperature of the lower surface of the hearth would be 600 F. The average temperature, therefore, is 1450 F, and from Fig. 90 the heat content is 368 Btu per lb. By the same method, for 1200 F inside, the steady-state temperature of the lower surface was found to be 378 F; average temperature 789 F; and heat content 179 Btu per lb.

Change of heat content

$$= \{28\%\} \times \left(9.75 \times 11 \times \frac{12\tfrac{1}{2}}{12}\right) \times 125 \times [368 - 179] = 740{,}000 \text{ Btu}$$

Note that the above volumes are based on inside-wall areas, and do not include edges. The quantity of heat taken from the edges during cooling, as mentioned previously, is 38 per cent of that abstracted from an equivalent weight of brick in the straight wall.

Edges. Top horizontal edges $(2 \times 11.25 + 2 \times 9.75) \times 0.75 \times 1.5 \times 125 = 5900$ lb

Change of heat content

$$= \{29\% \text{ average}\} \times 38\% \times 5900 \times [343 - 167] \text{ average} = 114{,}300 \text{ Btu}$$

Vertical edges: $4 \times 5.25 \times 1.5 \times 1.5 \times 125 = 5900$ lb

Change of heat content

$$= \{21\%\} \times 38\% \times 5900 \times [332 - 162] = 80{,}000 \text{ Btu}$$

Bottom horizontal: $(2 \times 11.25 + 2 \times 9.75) \times 1.5 \times \frac{(12\tfrac{1}{2})}{(12)} \times 125 = 8200$ lb

Change of heat content

$$= \{25\% \text{ average}\} \times 38\% \times 8200 \times [350 - 170 \text{ average}] = 147{,}500 \text{ Btu}$$

Total heat change in edges = 341,800 Btu

Corners: $\left(4 \times 1\tfrac{1}{2} \times 1\tfrac{1}{2} \times \tfrac{3}{4} + 4 \times 1\tfrac{1}{2} \times 1\tfrac{1}{2} \times \frac{12\tfrac{1}{2}}{12}\right) \times 125 = 2015$ lb

Change of heat content

$$= \{24\%\text{ average}\} \times 11\% \times 2015 \times [332 - 162\text{ average}] = 9580\text{ Btu}$$

The storage in the corners is so small that it will be neglected in the following.

Downtakes. Average temperature change in the inner surface of the two downtakes is from 1400 to 1800 F. Corresponding average temperatures throughout the brickwork of these walls, for steady-state conditions, are 851 and 1079 F, respectively.

Front and back walls:

Change of heat content

$$= \{27\%\} \times \left(2.25 \times 10 \times \frac{13\frac{1}{2}}{12} \times 125\right) \times 2 \times [257 - 196] = 104{,}300\text{ Btu}$$

Note that part of the back wall of the heating chamber and one wall of each downtake are built together, so that the temperature differences given above for ventilated walls, although correct for most of the area, are not quite correct for this part. More accurate calculation, using Schmidt's graphical method, showed so little error, however, that the use of the simpler approximate method is justified.

Sidewalls: Each 2 ft 3 in. by 10 ft 0 in. by 18 in.

Change of heat content

$$= \{21\%\} \times (2.25 \times 10 \times 1.5 \times 125) \times 2 \times [252 - 191] = 108{,}200\text{ Btu}$$

Roof: $2.25 \times 3.38 \times 0.75 \times 125 = 712\text{ lb}$

Change of heat content

$$= \{39\%\} \times 712 \times [338 - 163] = 48{,}700\text{ Btu}$$

Edges:

$$\left(4 \times 10 \times 1.5 \times \frac{13\frac{1}{2}}{12} + 2 \times 3.38 \times 1.125 \times 0.75 + 2.25 \times 1.125 \times 0.75\right) \times 125 = 9390\text{ lb}$$

Change of heat content

$$= \{25\%\text{ average}\} \times 38\% \times 9390 \times [254 - 193\text{ average}] = 54{,}400\text{ Btu}$$

Total heat-storage loss, per cycle, in downtakes

$$= 2 \times (104{,}300 + 108{,}200 + 48{,}700 + 54{,}400) = 631{,}200\text{ Btu}$$

Regenerators. The number of 9-in. straight brick used in each regenerator is 2400. The weight of the checkerwork is then

$$2 \times \frac{9 \times 4.5 \times 2.5}{1728} \times 125 \times 2400 = 35{,}170\text{ lb}$$

How much the average temperature of the checker bricks drops during the cooling period of 1½ hr depends on the operation of the damper. If the latter is closed at the end of heating while the ingot is being withdrawn, the stack will have cooled, and circulation may be reversed: that is, air may be drawn into the top of the stack and discharged through the open furnace door.

Usually, however, the stack damper remains open, and cold air is drawn in at the door and discharged through the stack. The cooling of the checkers then depends on how widely the damper is opened. On the basis of a downward velocity of air in the downtakes of 10 fps (referred to 60 F), and of practically constant temperature throughout the checker bricks (which, being thin, and heated and cooled from both sides, do not maintain any considerable internal temperature differences over this relatively long period), a calculation by graphical methods showed that the average temperature of the checkers would drop from 1500 to 1080 F during the cooling period. The corresponding change of heat content would then be

$$35{,}170 \text{ lb} \times [384 - 258] = 4{,}431{,}000 \text{ Btu}$$

Brick Walls Surrounding Regenerators.

Roof arches: Two arches 13 ft 6 in. by 5 ft 6 in. by 9 in. thick.

Change of heat content (for change from 1300 to 1700 F surface temperature)

$$= 2 \times \{39\%\} \times 13.5 \times 5.5 \times 0.75 \times 125 \times [250 - 188] = 336{,}500 \text{ Btu}$$

Sidewalls: Two side and one partition wall, each 13 ft 6 in. by 6 ft 3 in. by 13½ in. thick. Average temperature change of inner surface is from 1280 to 1050 F.

Change of heat content

$$= 3 \times \{27\%\} \times 13.5 \times 6.25 \times \frac{13\frac{1}{2}}{12} \times 125 \times [177 - 142] = 336{,}500 \text{ Btu}$$

Ends: Two ends, each 10 ft 6 in. by 6 ft 9 in. by 13½ in. thick.

Change of heat content

$$= 2 \times \{27\%\} \times 10.5 \times 6.75 \times 1.125 \times 125 \times [177 - 142] = 188{,}500 \text{ Btu}$$

Edges: Change of heat content $= 38\% \times \{33\% \text{ average}\} \times (4 \times 13.5 \times 1.125 \times 0.75 + 4 \times 10.7 \times 1.125 \times 0.75 + 4 \times 6.25 \times 1.125 \times 1.125) \times 125 \times [177 - 142 \text{ average}] = 62{,}300$ Btu.

Total change of heat content of walls surrounding checkers

$$= 336{,}500 + 336{,}500 + 188{,}500 + 62{,}300 = 923{,}800 \text{ Btu}$$

The total change of heat content of all the brickwork is the total of the separate items, or

Heating chamber....	arch	=	730,000 Btu
	sidewalls	=	685,000
	ends	=	828,700
	bottom	=	740,000
	edges	=	341,800
	corners	=	9,600
Downtakes.................		=	631,200
Checkerwork...............		=	4,431,400
Walls of regenerator chambers		=	923,800
Total.................		=	9,321,500 Btu

The charge in the furnace, for this heating cycle, is a steel ingot 26 in. in diameter projecting into the furnace 10 ft 0 in. The weight of the steel is

$$\frac{\pi}{4}(26)^2 \times 120 \times 0.283 = 18{,}050 \text{ lb}$$

The heat input to 1 lb of steel heated from 60 to 2270 F (average temperature of ingot) is 371 Btu (Fig. 27). The total heat absorbed by a charge of steel is

$$18{,}050 \times 371 = 6{,}696{,}000 \text{ Btu}$$

The ratio of useful heat to (useful heat plus heat absorbed in brickwork)

$$= \frac{6{,}696{,}000}{6{,}696{,}000 + 9{,}321{,}500} = 41.8\%$$

The result shows very plainly that much heat can be saved in this furnace if the ingots are preheated, in which case the furnace need not be cooled between ingots.

This example was selected, because it requires a great number of different, detailed calculations. However, a furnace of this type is not recommended for forge plants, if natural gas serves as fuel. The mixing of fuel and of preheated air is so poor that most of the combustion takes place in the regenerators. It is difficult to obtain a sufficiently high temperature in the heating chamber unless the natural gas is blown in by compressed air or by steam. In either case, the calculations have to be modified.

EXAMPLE 2. The second example is that of a simple oil- or gasfired furnace for heating rounds which are to be drop-forged. In this example, the material is put into the hot furnace; that is, the furnace is not cooled down during the day. The stock is heated to 2250 F. Flame temperature probably 2500 F. Inside wall temperature taken as 2350 F; gases leave furnace at 2400 F. Steel is actually heated during 8 hr per day, not including the heating-up time (which is about 2 hr). The furnace is shown in Fig. 137.

Heat Imparted to the Steel Being Heated. 75 lb of steel are heated per hr per sq ft of hearth. From Fig. 27, heat input to heat 1 lb of steel to 2250 F = 373 Btu.

$$(2\tfrac{5}{8} \times 5\tfrac{1}{4}) \times 75 \times 8 \times 373 = 3{,}083{,}000 \text{ Btu per day}$$

Losses from wall, including heat-storage loss: The wall loss is here computed by the same methods as in the fourth edition. At the end of the calculation for wall loss, it is recalculated by other methods, for the sake of comparison.

For the one-day heating cycle the radiation and convection loss from the walls and the loss by storage of unavailable heat in the walls are figured as a combined loss. The loss under steady-state conditions must first be calculated. The heat-conducting areas are taken halfway between outside and inside areas of walls (arithmetical mean).

4½-in. door: $\text{Area} = \left(\frac{\pi}{2} \times (16)^2 + 7 \times 32\right) \div 144 = 4.33 \text{ sq ft}$

From Fig. 86, heat loss when inside temperature equals 2350 F is 3370 Btu per (sq ft, hr).

$$4.33 \times 3370 = 14{,}600 \text{ Btu per hr}$$

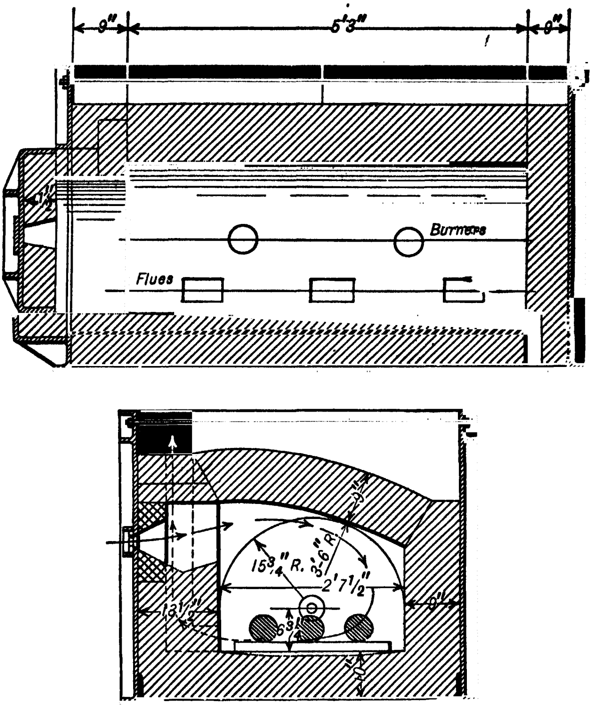


Fig. 137. Forge furnace.

9-in. roof and walls: The 13½-in. wall with flues may, with sufficient accuracy, be taken as equivalent to a 9-in. wall.

$$
\begin{array}{lrcl}
\text{Roof:} & \left(2\frac{5}{8} + \frac{3}{4}\right) \times 6 \text{ ft} & = & 20.2 \text{ sq ft} \\
\text{Sides:} & 2 \times \left(2 + \frac{3}{4}\right) \times 6 \text{ ft} & = & 33.0 \text{ sq ft} \\
\text{End:} & \left(2 + \frac{3}{4}\right) \times \left(2\frac{5}{8} + \frac{3}{4}\right) & = & 9.3 \text{ sq ft} \\
\text{End:} & 2\frac{3}{4} \times 3\frac{3}{8} - 4.33 & = & 5.0 \text{ sq ft} \\
\hline
 & \text{Total area} & = & 67.5 \text{ sq ft}
\end{array}
$$

1790 Btu per (sq ft, hr) [from Fig. 86] × 67.5 = 120,800 Btu per hr

10-in. hearth: The hearth is ventilated; it rests on I beams which permit air circulation (not shown in Fig. 137). The loss will be taken as 90 per cent of the loss from a freely exposed sidewall. Area (same as roof) = 20.2 sq ft. Interpolating in Fig. 86, the unit loss is 1660 Btu per (sq ft, hr).

$$90\% \times 1660 \times 20.2 = 30{,}160 \text{ Btu per hr}$$

The ratio of heat-storage loss plus loss by radiation and convection during the 8-hr working time plus the heating-up time, to the radiation and convection loss (only) in 8 hr if the steady state of heat flow were reached, is 2.35 for the 9-in. wall, 2.50 for the 10-in. hearth, and 1.5 for the 4½-in. door.

2.35 × 120,800 × 8 = 2,270,000 Btu per day
2.50 × 30,160 × 8 = 603,200 Btu per day
1.50 × 14,600 × 8 = 175,200 Btu per day
Total = 3,048,400 Btu per day

This is the total loss by heat storage in the brickwork as well as by conduction through and radiation and convection from the walls. It is based on operation during 5 days per week. (If the furnace is operated fewer days per week, a correction is required.)

Effect of 8-hr-per-day operation on wall loss in accordance with Fig. 91: In this method, the steady-state heat loss in 24 hr is multiplied by a coefficient which is smaller than unity, whereas in the method previously followed, the heat loss in 8 hr was multiplied by a coefficient greater than unity

Roof and walls	120,800 × 24 × 0.61	1,768,500 Btu per day
Hearth	30,160 × 24 × 0.63	456,000 Btu per day
Door	14,600 × 24 × 0.43	150,700 Btu per day
	Total	2,375,200 Btu per 24 hr

This value is about 22 per cent below the above determined figure. From a purely theoretical viewpoint, a difference of 22 per cent is excessive. Practically, it is tolerable. The chart, Fig. 91, neglects several factors for the sake of simplicity. Furthermore, the effect of leakage through burner openings and walls is uncertain.

Wall loss, if wall and roof, but not the hearth, are built of 2800 F lightweight firebrick: From Fig. 87, the hourly heat loss through 1 sq ft of 4½-in. brick is 1320 Btu, for a furnace temperature of 2350 F. The corresponding value for 9-in. bricks is 680 Btu. With the heat-conducting area halfway between outside and inside areas of walls, the hourly heat loss equals 4.33 × 1320 + 67.5 × 680 + 0.9 × 1660 × 20.2 = 81,920 Btu per hr. This is not much more than one half of the heat loss that is incurred with walls of dense firebrick.

Radiation through Door Opening. The door is open to 6-in. height for 50 per cent of the time. Ratio of sides of rectangle = 2⅝: ½ = 5.25. Ratio of width of opening to thickness of wall = 6 in./9 in. = 0.67. Interpolating in Fig. 96, the total radiation factor for an opening of these proportions is 0.55, approximately. From Fig. 95, blackbody radiation at 2350 F is 745 Btu per (sq in., hr).

$$745 \times 6\text{ in.} \times 31\tfrac{1}{2}\text{ in.} \times 0.55 \times 8 \times 50\% = 310{,}000\text{ Btu per day}$$

Loss of Furnace Gases around Door and through Opening. ½-in. crack around the door.

Crack: $$\frac{(\pi \times 16 + 2 \times 7) \times \frac{1}{2}}{144} = 0.222\text{ sq ft}$$

Door itself, when open: $$\frac{6 \times 32}{144} \times 50\% = 0.667\text{ sq ft}$$

Total area = 0.889 sq ft

Density of gases at 2400 F = 0.0134 pcf or specific volume = 74.5 cu ft per lb = v

Let the excess pressure in the furnace be $P_1 - P_2 = 0.025$ in. of water, or 0.13 psf, and coefficient of discharge be 0.75.

Velocity of gases escaping

$$= \sqrt{2g \times (P_1 - P_2) \times v} = \sqrt{64.4 \times 0.130 \times 74.5} = 25 \text{ fps}$$

Weight of gases escaping

$$= \frac{0.889 \text{ sq ft} \times 25 \times 3600 \times 0.75}{74.5} = 805 \text{ lb per hr}$$

Since 15.07 lb of products are formed per lb of fuel oil, the weight of fuel used to produce the escaping gases = 805/15.07 = 53.4 lb per hr or 428 lb per day.

Fuel Required. Gases leave the furnace at 2400 F during the 8-hr operating period. During the heating-up period of about 2 hr, their average leaving temperature is estimated to be 2000 F.

Let the combustion device be so adjusted that sufficient air is supplied for perfect combustion, but let the waste gases contain 1 per cent CO.

Lower heating value of oil = 18,830 Btu per lb.[*]

Sensible heat in products of perfect combustion at 2400 F = 10,100 Btu per lb of oil

Heat lost by incomplete combustion = 1000 Btu per lb of oil

Heat transferred in the furnace by the products of combustion of 1 lb of oil, 2400 F leaving temperature, = 18,830 − 10,000 − 1000 = 7730 Btu

During the heating-up period, the heat in the products leaving at an average temperature of 2000 F is 8200 Btu per lb of oil.

$$18{,}830 - 8200 - 1000 = 9630 \text{ Btu}$$

$$\text{Average amount of heat utilized per pound of oil} = \frac{8 \times 7730 + 2 \times 9630}{8 + 2}$$

$$= 8110 \text{ Btu}$$

Heated transferred to stock, per day = 3,083,000 Btu
Heated stored in brickwork and lost through walls, per day = 3,048,400 Btu
Heat lost by radiation through openings, per day = 310,000 Btu
Total = 6,441,400 Btu

Let the gases which escape around the door contain 2 per cent CO + 2 per cent H_2. Then the loss by incomplete combustion = 2060 Btu per lb oil. Hence these gases, at 2000 F, give up in the furnace 9110 − 2060 = 7050 Btu per lb oil, or 7050 × 428 = 3,017,000 Btu per day.

Heat remaining to be furnished by the gases which pass out through the flues = 6,441,400 − 3,017,000 = 3,424,400 Btu per day

$$\text{Fuel consumption} = \frac{3{,}424{,}400}{8110} + 428 = 850 \text{ lb per day, or 126 gal per day}$$

Overall thermal efficiency[†] of the furnace

$$\frac{3{,}083{,}000 \text{ Btu heat in steel}}{850 \times 19{,}980 \text{ higher heating value}} \quad 0.182, \text{ or } 18.2\%$$

[*] This example was figured on the basis of burning no. 5 fuel oil. For that reason the heat contents etc. differ slightly from those of Fig. 112 which contains information on no. 6 fuel oil.

[†] In American practice, efficiencies are based on the higher or gross heating value, instead of the more logical lower or net heating value.

Fuel used per net ton of steel heated

$$\frac{850 \times 2000}{(2\tfrac{5}{8} \times 5\tfrac{1}{4} \times 75 \times 8 = 8270 \text{ lb per day})} = 205 \text{ lb of oil}$$

or 30.5 gal of oil per net ton of steel.

HEAT BALANCE

	Btu per day	Per cent
Heat input to steel (useful heat)	3,083,000	18.2
Wall losses (storage plus conduction)	3,048,400	17.9
Heat radiated through openings	310,000	1.8
Heat lost in unburned combustible in fuel gases	882,000	5.2
(escaping through openings)	422,000	2.5
Sensible heat lost in products of combustion	8,259,600	48.6
Latent heat of water vapor, not recovered	978,000	5.8
Total = heat in fuel	16,983,000	100.0

It must be understood that considerable variation from these figures may occur in practice, because conditions may not be the same as those stated. After a furnace has been in operation for some time, the roof burns thinner, the walls become thinner, and cracks open in them, permitting gases to escape and heat to be lost by radiation. Then there are the factors that depend on the operator and on the type of fuel-burning equipment. If the combustion device is improperly designed or operated, there will be an additional loss from unburned combustible escaping with the stack gases, and less of the potential heat of the fuel will pass into the steel. If the operator leaves the door open all the time, the third and fourth items in the table will be greatly increased. The efficiency figured is not likely to be increased, but much lower efficiencies may be encountered.

Heat developed by oxidation of the steel has not been allowed for in this example, because, in well-designed and properly operated furnaces, the oxidation is small.

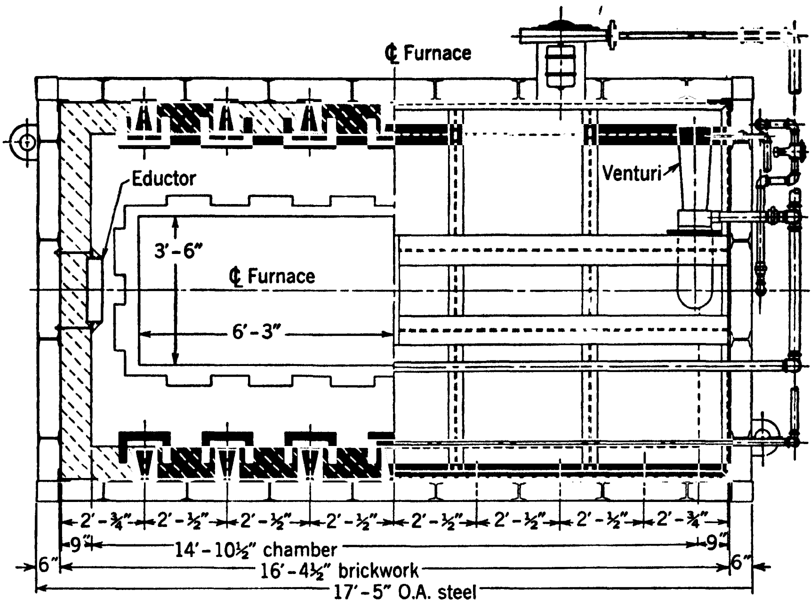
EXAMPLE 3. 63,500 lb of steel sheets are to be heated to 1300 F in the hood-type furnace which is illustrated by Fig. 138. After having been heated and soaked, the sheets will stay in the box until they have been cooled down to a temperature of a few hundred degrees. Natural gas is to be the fuel. What is the fuel consumption per cycle?

The following quantities must be computed:

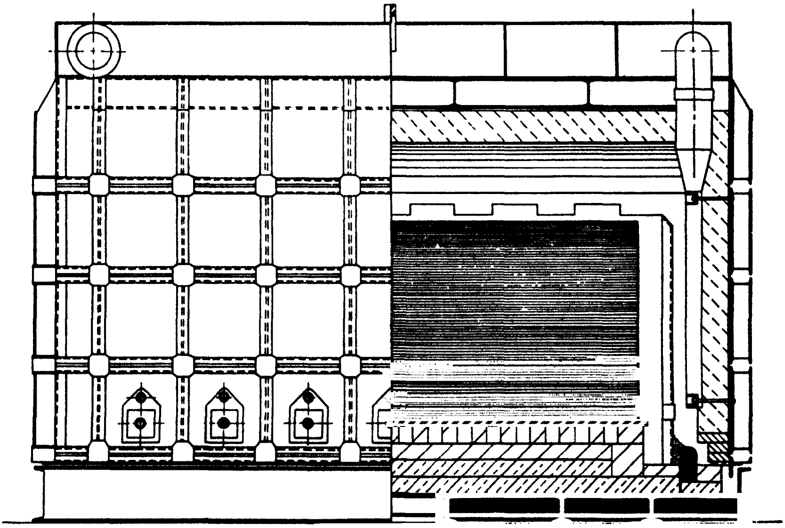
1. Heat imparted to sheets.
2. Heat stored in annealing box.
3. Heat stored in base under sheets.
4. Heat imparted to bricks and ground under base.
5. Heat imparted to protective gas.
6. Heat stored in hood.
7. Heat given up by outside of hood.
8. Heat lost in products of combustion.

Items 1, 2, 3, 4, and 6 can be computed directly from physical data shown on the drawing. For the calculation of items 5, 7, and 8, the heating time must be computed. Calculation of the various items follows.

1. From Fig. 27, the heat required to raise the temperature of 1 lb of steel

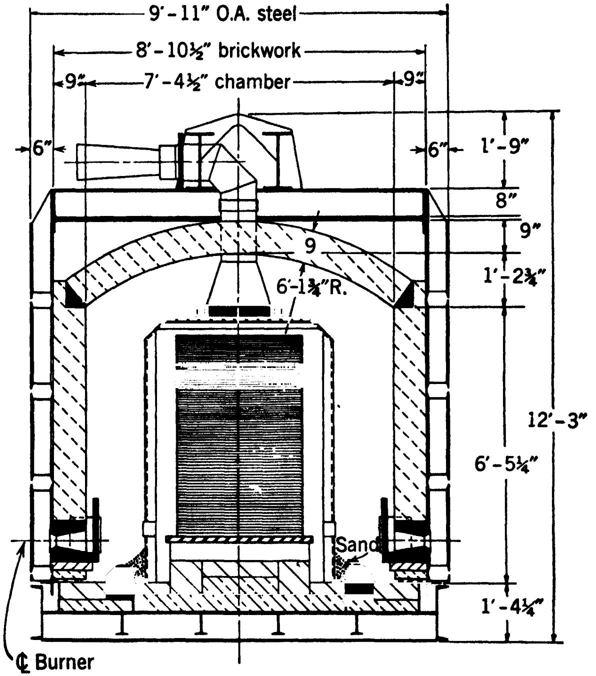


Half Plan Section Half Plan



Half Side Elevation Half Longitudinal Section

Fig. 138 (*a*). Hood-type annealing furnace.



Cross Section

Fig. 138 (*b*). Hood-type annealing furnace.

from 70 to 1300 F equals 2200 − 8 = 192 Btu. Heat imparted to sheets then equals 63,500 × 192 = 12,200,000 Btu.

2. Weight of annealing box equals 4140 lb. Since it transmits heat to the sheets, its temperature must exceed that of the sheets. Final temperature of box (average with regard to space) is estimated to be 1350 F. Heat per pound, from Fig. 27, equals 210 − 8 = 202 Btu. Heat stored in annealing box = 4140 × 202 = 830,000 Btu.

3. Weight of metallic base under sheets equals 18,000 lb. The base has a slightly higher temperature than that of the sheets in the bottom layers, because hot deoxidizing gas passes through the hollow base. A temperature of 1320 F is used in this calculation. Then, heat per pound equals 205 − 8 = 197 Btu, and heat stored in metallic base equals 18,000 × 197 = 3,550,000 Btu.

4. The heat to be imparted to bricks and ground under the base varies with the temperatures that the bricks and the ground below them have at the beginning of the heating cycle. On account of this uncertainty, it is close enough to assume that a layer of bricks 1½ ft thick is to be heated to a temperature of 1200 F. Then, heat imparted to subbase equals

$$\overset{\text{Floorspace}}{4\tfrac{1}{2}\text{ ft} \times 13.7\text{ ft}} \times \overset{\text{Thickness}}{1.5\text{ ft}} \times \overset{\text{Temperature rise}}{(1200\text{ F} - 100\text{ F})} \times 130\text{ pcf} \times \overset{\text{Specific heat}}{0.23}$$

This equals 3,100,000 Btu.

5. The heat to be imparted to the deoxidizing gas is always small, almost

negligible compared to the other heat quantities. It varies with the tightness of the seal under the annealing box. It is estimated that 450 cu ft (33.8 lb) of gas are passed through the annealing box in 1 cycle. The specific heat of the gas is 0.3; its average temperature rise is 1200 F. Then, the heat to be imparted to the gas equals approximately 12,000 Btu, which is indeed small compared to the other heat quantities.

6. Heat stored in hood (outer cover) equals weight of bricks times specific heat times temperature rise. Volume of brickwork is computed as follows:

Two sidewalls	145 cu ft
Two end walls	86 cu ft
Roof	204 cu ft
Bottom ring, outside of sand seal	40 cu ft
Total	475 cu ft

The cover lining consists of refractory insulating brick which weighs approximately 80 pcf. In order to transmit enough (but not too much) heat to the annealing cover, the inner wall surfaces of the hood must have an average (with regard to space) temperature between 1450 and 1500 F. A temperature of 1470 F is used here. From Fig. 85, the outside temperature equals 340 F, because of the metal casing. The average wall temperature at the end of the heating period then is $\frac{1}{2}(1470 + 340) = 905$ F. The heat stored in the cover equals

$$475 \text{ cu ft} \times 80 \text{ pcf} \times \overset{\text{Specific heat}}{0.23} \times \overset{\text{Temperature rise}}{(905 \text{ F} - 70 \text{ F})} = 7{,}350{,}000 \text{ Btu}$$

If the hood, while still hot from the preceding heat, is placed over a new charge, then the quantity of heat to be stored in the hood is, of course, smaller.

7. Before the heat transmitted through the hood and dissipated by it can be found, the heating time must be calculated. The sheets are 3 ft wide, which means that heat must penetrate 18 in. from either long side to the center. The rule that heat penetrates ⅛ in. in 5 min would furnish a heating time of 12 hr. That rule can, however, not be applied here, for several reasons. The rule is based on the assumption that the charge be placed in a furnace which is already up to its final temperature. In the sheet-annealing furnace, much of the initial heat of combustion goes to heating the hood and the annealing box. Furthermore, the corners of the sheets receive heat from two sides. Heating must proceed so slowly that the corners are not overheated. And, finally, the top layers of the sheets receive heat from above. Heating must progress slowly enough to afford time for downward flow of heat from the top sheets without overheating the top sheets. Downward flow of heat is slow, because the sheets in the top layers contact in spots only. All of these heat flows can be determined mathematically, if the time and the money are available. Trial runs on similar furnaces have shown, without long calculations, that a penetration of ⅛ in. in 10 min results in uniform heating. For this example, the rule furnishes a heating time of 24 hr, which will be used in the calculations. It may be noted that the heating time can be shortened, if the top of the annealing box is insulated.

For the above given reasons (slow heating), the average (with regard to space) temperature of the inside surface of the hood must be comparatively low, never exceeding the final temperature of the sheets by more than 275 to

300 F. With regard to time, the inside temperature of the hood gradually rises to a maximum, and then fluctuates up and down, under the action of the automatic temperature control. The time average is estimated to be 1500 F. Near the beginning of the heating cycle, very little heat passes through and out of the hood, because the greater part of the heat flowing into the hood is stored in the wall. This heat quantity was computed in item 6. For this reason, heat dissipation from the walls is estimated to take place during 20 hr only.

The heat-dissipating surfaces are: Sidewalls, 193 sq ft; end walls, 115 sq ft; roof, 132 sq ft; bottom ring, 59 sq ft. The lining of the hood consists of 2300 F refractory insulating brick, except at and near the burners, where a more refractory material is needed. The roof is covered with insulation of varying thickness. From Fig. 87, the hourly wall loss for 2300 F brick and for 1500 F inside temperature is 280 Btu per sq ft. For the insulated roof, the corresponding value is about 200 Btu per sq ft. The total heat dissipation per cycle then equals $20(193 + 115 + 59) \times 280 + 20 \times 132 \times 200 = 2{,}583{,}000$ Btu.

8. Heat lost in outgoing products of combustion. The temperature of the products of combustion, at the point where they leave the heating chamber, can differ but little from the wall temperature of the hood. Hence, a temperature of 1480 F is used as a time average for calculation. It is assumed that the gas will be burned with 5 per cent excess air. Then, from Fig. 114, the heat carried out of the furnace per cubic foot of fuel gas equals $355 + 0.05 \times 295 = 370$ Btu. Since the lower heating value equals 993 Btu per cu ft, the difference, or roughly 623 Btu per cu ft, is available for heating. The gas consumption per cycle is the sum of all heat expenditures divided by 623, or $29{,}625{,}000/623 = 47{,}550$ cu ft. Dividing this figure by the tons of sheets in the charge and multiplying by the local gas rate furnishes the fuel cost of annealing a ton of sheets.

Although several "guesstimates" were made in the calculation, comparison with runs in which the fuel consumption was measured shows that the result of the calculation is accurate within 3 or 4 per cent.

EXAMPLE 4. A furnace of the design shown in Fig. 139 with internal recirculating fan and alloy baffles is arranged with a conveyor comprised of two strands of alloy chain and connecting slats. The conveyor passes over sheaves at each end and returns under the furnace. Aluminum billets are carried by the conveyor and are heated at the rate of 2500 lb per hr to 925 F. The weight of the conveyor passing through the furnace per hour is 1000 lb. The temperature of the gases recirculating in the furnace is 960 F, and the average drop in temperature of the gases is 100 F. Cross section of the furnace is shown in Fig. 139, and the length of the furnace is 22 ft. Fuel is natural gas.

Heat in Aluminum. From Fig. 26, heat input to raise 1 lb of aluminum from 60 F to 925 F is 212 Btu.

$$2500 \text{ lb per hr} \times 212 = 530{,}000 \text{ Btu per hr.}$$

Heat in Conveyor. The conveyor returns below the furnace, and the time of exposure to the air is more than 1½ hrs, so that the heat remaining at the time the furnace conveyor re-enters the charging end can be neglected. From Fig. 27, the heat to the conveyor is 1000 lb per hr × 110 Btu per lb = 110,000 Btu per hr.

Heat Loss by Conduction through Walls. It is assumed that the furnace operation is continuous throughout the week, and the heat absorption by refractories when starting up is neglected. The mean temperature of the hot face of the walls along the length of the furnace is taken as 900 F, and the average conducting area, including edges, is taken as the center of the walls, bottom, and arch.

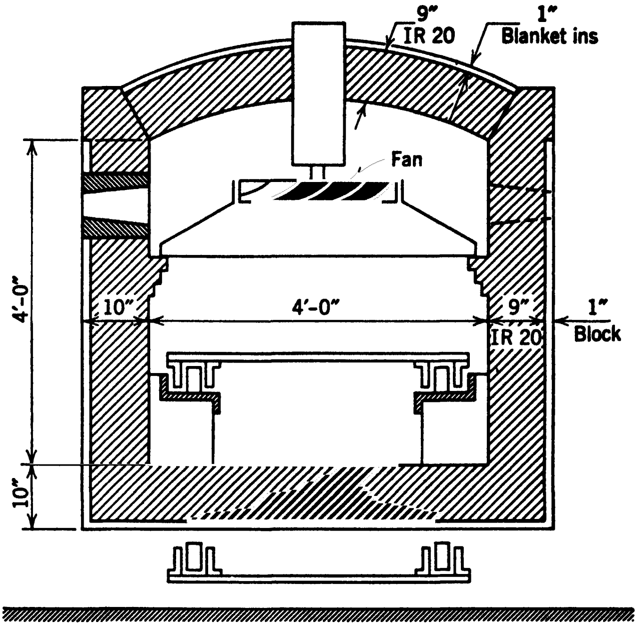


Fig. 139. Cross section of slat conveyor furnace with recirculating fan for heating aluminum billets.

9-in. insulating refractory IR20 and 1-in. block insulation walls and bottom at 900 F:

$$3 \times 4\tfrac{5}{6} \text{ ft} \times 22\tfrac{5}{6} \text{ ft} = 329 \text{ sq ft}$$

$$329 \times 90 \text{ interpolated from Fig. 87} = 29{,}500 \text{ Btu per hr}$$

9-in. and 1-in. arch at 900 F:

$$2\pi \times \frac{4\tfrac{5}{6}}{6} \times 22\tfrac{5}{6} = 115 \text{ sq ft}$$

$$115 \times 90 = 10{,}350 \text{ Btu per hr}$$

4½-in. doors at 900 F:

$$2 \times 4 \times 2 = 16 \text{ sq ft (interpolated from Fig. 87)}$$

$$16 \times 180 = 2890 \text{ Btu per hr}$$

9-in. insulating refractory IR20 and 1-in. block end walls:

$$45 \times 90 = 4050 \text{ Btu per hr}$$

Total heat through walls:

$$29{,}500 + 10{,}350 + 2890 + 4050 = 46{,}790 \text{ Btu per hr}$$

Radiation through Door Openings. The doors at both ends are allowed to be open for a height of 12 in. to permit continuous passage of conveyor and stock in and out of the furnace.

Ratio of length to width of opening = 48/12 = 4.0

Ratio of width of opening to thickness of wall = 12/10 = 1.2

From Fig. 96, the radiation factor by interpolation is 0.70

From Fig. 95, the blackbody radiation at 900 F = 40 Btu per (sq in. hr)

$$2 \times 12 \times 40 \times 48 \times 0.70 = 32{,}200 \text{ Btu per hr}$$

Heat in Flue Gases. From Fig. 114:

Higher heating value = 1097 Btu per cu ft of gas at 60 F

Lower heating value = 993 Btu per cu ft of gas at 60 F

Sensible heat in products of combustion of 1 cu ft of natural gas, plus 15 per cent

Excess air at 960 F (Fig. 114) = 220 + 0.15 × 180 = 247 Btu

In a furnace of this kind, combustion can be considered to be complete.

Heat given up by gases in the furnace, per cu ft of fuel = 993 − 247 = 746 Btu

Total heat that must be given up by the gases in the furnace = 530,000 to aluminum + 110,000 to conveyor + 46,790 wall loss + 32,000 to radiated through openings = 721,990 Btu per hr.

$$\text{Fuel consumption} = \frac{721{,}990}{746} = 970 \text{ cu ft per hr}$$

$$\text{Fuel used per lb of aluminum} = \frac{970}{2500} = 0.39 \text{ cu ft}$$

This is for operation on a 24-hr continuous basis, neglecting heating-up fuel over week-ends. A monthly average including week-ends and usual variations will be about 0.51 Btu per lb.

15. DATA ON FUEL CONSUMPTION OF VARIOUS TYPES OF FURNACES

The consumption of potential heat energy by furnaces varies between wide limits with the design of the furnace, with the form of the fuel, and with the operation of the furnace. For this reason it is somewhat risky to give fuel-consumption data, and experienced furnace engineers frown on such data as being misleading.

Nevertheless, these same furnace engineers carry in their heads certain average values to which they, consciously or unconsciously, compare the fuel consumptions of furnaces coming under their observation. In Table 16 are given such average values, **with the understanding that they are only averages and that the actual fuel consumption of a given furnace may depart considerably from the**

TABLE 16. Typical Heat Consumptions of Industrial Furnaces

Heating for Process	Material to Be Heated and Approximate Temperature	Type of Furnace	Heat Consumption, Btu per Net Ton Heated	
			Average	Minimum Recorded
Rolling	Steel, 2150 to 2250 F	Continuous, no heat salvage	2,200,000	1,600,000
Rolling	Steel, 2150 to 2250 F	Continuous, no heat salvage, hot charge	1,800,000	1,400,000
Rolling	Steel, 2150 to 2250 F	Continuous, recuperator	1,600,000	1,000,000
Rolling	Steel, 2150 to 2250 F	Continuous, regenerator	1,650,000	1,100,000
Rolling	Steel, 2150 to 2250 F	In and out, no heat salvage	4,400,000	2,200,000
Rolling	Steel, 2300 F	In and out, regenerative	3,100,000	1,600,000
Rolling	Steel, 2100 to 2350 F	Rolldown, regenerative or recuperative	1,900,000	1,400,000
Rolling	Steel, 2250	Axial	3,750,000	
Rolling	Steel, 2150 to 2250 F	Pit furnace, hot charge	900,000	330,000
Rolling	Steel, 2150 to 2250 F	Same, cold charge	3,800,000	2,800,000
Rolling	Stainless steel, 2400 F	In and out, recuperative	4,200,000	3,400,000
Forging	Steel, 2150 to 2250 F	In and out, no heat salvage	7,000,000	2,500,000
Forging	Steel, 2150 to 2250 F	In and out, recuperator	4,500,000	2,000,000
Forging	Steel, 2150 to 2250 F	Rotating hearth	3,000,000	2,600,000
Forging	Steel, 2150 to 2250 F	Continuous	2,800,000	
Forging	60" diam. steel ingot	Batch type 14-hr heating time	23,000,000	
Drop forging	Steel, 2100 to 2500 F	In and out, no heat salvage	8,000,000	5,000,000
Drop forging	Steel, 2100 to 2500 F	In and out, recuperator	4,500,000	2,800,000
Welding	Steel skelp, 2500 F	In and out, regenerator	5,000,000	2,600,000
Welding	Steel skelp, 2500 F	Axial	4,500,000	2,800,000
Annealing	Steel castings	Car type	3,000,000	1,800,000
Annealing	Steel	Continuous, counterflow	1,500,000	1,000,000*
Annealing	Steel sheets	Box annealing	5,000,000	3,000,000†
Annealing	Steel sheets, 1300 F	Box annealing, outer cover lightweight brick	2,200,000	
Annealing	Tin-plate coils, 1300 F	Outer hood of lightweight refractory bricks	980,000	

TABLE 16—*Continued*

Heating for Process	Material to Be Heated and Approximate Temperature	Type of Furnace	Heat Consumption, Btu per Net Ton Heated	
			Average	Minimum Recorded
Annealing	Steel sheets, 1500 F	Continuous, on rollers and on wasters	1,900,000	1,000,000
Annealing	Steel wire	Pot furnace	4,200,000	1,700,000
Annealing	Steel wire	Continuous, varies with final temperature	4,000,000	1,200,000
Lead-bath patenting	Steel wire	Continuous, includes lead bath	5,500,000	
Rolling	Steel, 1500 F	Sheet and pair furnaces	4,500,000	
Normalizing	Steel sheets (auto body)	Various types of furnaces, varies with temperature	3,400,000	1,800,000‡
Normalizing	Steel sheets, stainless	Continuous	5,000,000	‡
Normalizing	Steel tubes	Continuous	2,200,000	1,200,000§
Quenching	Steel	Batch type	2,600,000	
Quenching	Steel	Continuous	1,400,000	
Quenching	Steel	Axial	2,600,000	
Drawing or tempering	Steel	Batch type, continuous	1,500,000 700,000	600,000
Drawing or tempering	Steel	Axial	1,200,000	
Carburizing (pack hardening)	Steel	Batch type, continuous, counterflow	8,000,000 3,800,000	‖ 2,000,000¶
Enameling	Steel	Muffle, batch type	9,000,000	8,000,000**
Enameling	Steel, ground coat	Continuous	4,500,000	††
Enameling	Steel, finish coat	Continuous	3,300,000	††
Rolling	Brass	Continuous, pusher type	900,000	700,000
Rolling	Copper	Continuous, pusher type	1,100,000	900,000
Annealing	Brass	Batch type	1,400,000	1,100,000
Annealing	Brass	Electric, roller hearth		300,000‡‡
Annealing	Copper	Box annealing, bright	5,500,000	
Annealing	Copper	Continuous bright annealing	900,000	
Tinning	Tin plate	Hot-dip process	850,000	

Figures are given in Btu per net ton of material heated (and *not* per ton of stock plus container, if any). Most of the data are from the experience of the senior author. Others were furnished by M. H. Mawhinney and by the W. S. Rockwell Company. Some were taken from the "Anhaltszahlen."

* 2 : 1 ratio of steel to trays.

† 1 : 1 ratio of sheets to (box plus cover).

‡ 1 : 1 ratio of sheets to riders.

§ rolled on sloping hearth, without containers.

‖ 1 : 2 ratio of steel to (compound plus container).

¶ 1 : 1½ ratio of steel to (compound plus container).

** 1 : 3 ratio of ware to tools.

†† 1 : 1 ratio of ware to tools.

‡‡ 10 : 1 ratio of brass to pan weight.

figures in the table. The lowest fuel consumption will seldom go below 60 per cent of the given values; the highest may exceed the average values by 100 per cent, or even more in extreme cases.

EXAMPLE. The oil consumption of a simple ingot-heating furnace in a forge plant is 4,500,000/135,000, which equals approximately 33 gal per net ton. The denominator in the fraction is the heating value of the oil (see Volume II, Table III). The rate of 33 gal per ton is a good average value, but oil consumptions up to 90 gal per ton have been found in flagrant cases.

chapter 6

Heat Saving Methods and Apparatus with Particular Reference to Industrial Furnaces

1. GENERAL STATEMENTS ON HEAT SAVING

In many industrial heating processes, fuel represents only a very small fraction of the total cost of manufacturing. In such cases, fuel saving is frequently considered as being not worth while. But there are other processes in which fuel represents a considerable item of expense. Although fuel is comparatively cheap in the United States, its cost is continually rising. For about fifty years, the cost of coal rose about 4 per cent compounded each year. Since about 1940, the rise in cost has been greatly accelerated. Although the rise in cost of oil and natural gas is less spectacular, it is, nevertheless, very noticeable. The cost of electric energy also rises because of the increase in the cost of coal, higher wages, and higher costs of equipment. In times of a buyers' market, competition among manufacturers becomes so keen that the difference between fuel saving and fuel wasting often determines the difference between profit and loss. For that reason, heat-saving methods must be given consideration.

Heat may be saved by the use of one or more of the following methods:

(*a*) Reduction of wall losses, including storage loss.

(*b*) Application of high-temperature heat to a low-temperature charge without loss.

(*c*) Preheating of the stock by waste heat.

(*d*) Preheating of air or fuel (or both) by waste heat.

(*e*) Waste heat boilers.

2. REDUCTION OF WALL LOSSES

As mentioned in Chapter 5, flaming out from poorly fitting doors is part of the wall loss. Well fitting doors are illustrated in Chapter 7.

Furnace walls built of insulating refractories and encased in a steel shell reduce flow of heat to the surroundings. The loss is still further reduced by the insertion of fiber block between the insulating refractory and the steel casing.

The walls of tall furnaces are often built of dense firebricks. The question then arises: How much can the heat loss be reduced by the application of insulation? The answer depends on thicknesses of firebrick and of insulation, on type of insulation, and on continuity of furnace operation. The manner in which the saving in heat varies with three of these variables can be seen from Table 17, which refers to wall losses only and not to the total heat consumption of the furnace.

TABLE 17. Reduction of Wall Losses by Insulation, Per Cent

Does not Include Edges: See Below

Thickness of of Firebrick Wall in.	Continuous operation		1-week Cycle, 2½-in. Insulation	1-day Cycle 6 days per week 2½-in. Insulation
	2½-in. Insulation	5-in. Insulation		
4½	62	76	58	25
9	46	65	36	18
13½	38	57	20	14
18	35	53	15	12

The permissible thickness of insulation, in combination with thickness of firebricks is given in Chapter 7. One-week cycle means continuous operation for 6 days of 24 hr. For 5-day operation, the saving is reduced approximately 10 per cent. One-day cycle means 8 to

10 hr per day. The tabular values must be reduced somewhat, if the wall is thick in comparison with the interior dimensions of the furnace. The tabular values apply to those furnaces only which are covered entirely with insulation, even under the binding.

Saving of heat does not necessarily mean saving of money, because the fixed charges on the cost of insulation may exceed the cost of the fuel that is saved. Although this is seldom the case, it must be taken into consideration.

Another factor that reduces the profitableness of insulation is its application to walls that are subject to frequent repairs. Examples are furnaces near steam hammers and furnaces that are heated up too quickly after a prolonged shutdown. In such furnaces, spalling occurs. None of the originally installed insulation can be salvaged when extensive repairs are made.

3. HEAT SAVING IN LOW-TEMPERATURE FURNACES AND OVENS

The flame temperature of any fuel must be reduced before the products of combustion contact the charge. Fulfillment of this command presents no difficulty in high-temperature furnaces. But if the stock is to be heated to temperatures between 800 F and 1300 F, finding a good solution is more difficult. The temperature of products of combustion is often reduced by admixture of excess air; the flame is "tempered." This frequently used method is convenient but wasteful, because the waste gases carry out not only the heat in the products of combustion but also the heat in the excess air.

In low-temperature furnaces, fuel is saved, if the products of combustion transfer part of their heat to the charge by radiation before contacting the charge. This principle has been successfully applied in the petroleum industry. A flame located in the center of a large furnace radiates to pipes that almost cover the surrounding walls. The partly cooled gases then contact other pipes. In industrial (metal-heating) furnaces it was an indifferent success only in the heating of sheets. The method, which is now obsolete, requires a tall furnace with heat release immediately under the roof. Most of the heat saved by this method is lost by increased wall loss. Furthermore, tall furnaces are expensive.

The best method of saving heat in low-temperature furnaces consists in mixing the hot products of combustion with cooler products that have already passed over the charge. Mixing may be accomplished by the jet action of the flame, as in Fig. 7 or by a hot fan as in Fig. 10. The method is fully discussed in Chapter 3. Jet action is discussed in Chapter 8.

4. UTILIZATION OF HEAT IN FLUE GASES

The loss caused by sensible heat in the flue gases (the stack loss) is illustrated by Fig. 140. The curves are based on fuel and air entering the burner at room temperature and also on a small percentage of excess air. At high temperatures, the loss becomes excessive. The desire to reduce the stack loss led to three heat salvaging methods:

(1) Preheating of cold stock by the flue gases.
(2) Preheating of combustion air (and of some fuels).
(3) Steam generation in waste heat boilers.

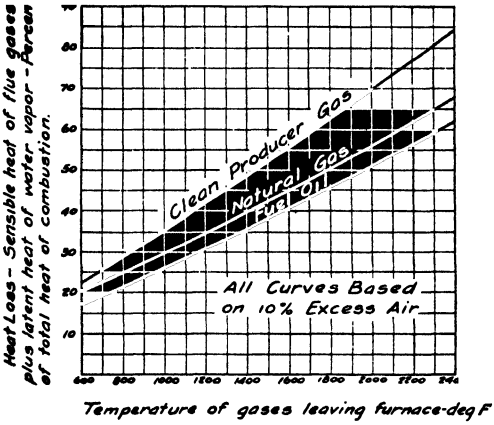


Fig. 140. Stack loss as a function of flue temperature and of form of fuel.

Preheating of cold stock, method (1), was first used in preheating chambers, later in some continuous furnaces. If steel is heated for forging or rolling, preheating the work blanks in a separate furnace by means of the waste gases of the high-heat furnace theoretically saves 30 per cent of the fuel. Nevertheless, this method is not practiced in the United States, because of first cost and of cost of labor.

(a) Preheating of Stock. Preheating furnaces (usually preheating pits) are in use for preheating ingots or large forging billets of tender steels. Such pieces are heated slowly to prevent cracking of the ingots. However, the pits are fired separately, and not by waste gases from the high-temperature furnace. Separate slow heating permits better control of temperature than heating by waste gases.

Preheating of stock by waste gases is widely practiced in the forging and hardening of tools. A furnace serving this purpose is illustrated in Fig. 17. Its use is limited to the heating of pieces that can be lifted and handled by a man.

Theoretically, a continuous furnace of the pusher type or of the doughnut type can be so operated that it saves fuel. If the rate of heating in pounds per (sq ft, hr) is low, most of the heat is imparted to the stock at the hot end, and the products of combustion can preheat the slowly moving work blanks. This simple theory is upset by heat losses through walls and to water-cooled parts and, above all, by investment costs and labor costs. The decision whether or not it pays to operate a continuous furnace as a fuel saver (by preheating the entering stock) depends on many circumstances, such as cost of fuel, return on the investment, floor space, permissible rate of heating, temperature uniformity in heated stock and still others. The calculations given in Chapter 5 are helpful in this connection.

(b) Fuel Saving by Preheating of Combustion Air. The second method of improving the economy of combustion-type furnaces consists in utilizing the heat of the stack gases for the preheating of combustion air, of fuel, or of both.

In connection with this method, three questions are of practical interest, namely: (1) What fraction of the fuel can be saved by the preheating, to a given temperature, of either the fuel or the air? (2) What type and what area of heating surface are required to secure the desired preheat? (3) Under what conditions is it advisable to use preheating devices? Answers to the first two questions are given in the present chapter; an answer to the third question is given in Volume II.

The fraction of the fuel that can be saved by the preheating of the combustion air can easily be computed by a method which will become clear from a study of Fig. 141. In that diagram, $Q + H$ = heat produced by combustion of unit quantity of fuel; Q = heat carried off by the products of combustion, including 5 per cent excess air (per unit quantity of fuel), while H = heat left in the furnace by the products of combustion. The higher the temperature of the flue gas, the greater Q and the smaller H. For the preheating of 105 per cent of that quantity of air which is needed for the combustion of unit quantity of fuel to a given temperature, P heat units (see illustration) are needed. It is evident that H is the heat utilized when there is no preheating, and that $H + P$ is the heat utilized when there is preheating. Hence, the fuel consumptions are proportional to $1/H$ and $1/(H + P)$, while the relative fuel saving (saving divided by the quantity used before saving) equals

$$\frac{\dfrac{1}{H} - \dfrac{1}{H+P}}{\dfrac{1}{H}} = 1 - \frac{H}{H+P} = \frac{P}{H+P} = \frac{1}{\dfrac{H}{P} + 1} \qquad (22)$$

It should be noted that Fig. 141 and equation 22 are based on preheat by the flue gases only, and do not include preheating by passing the air along hot furnace walls.

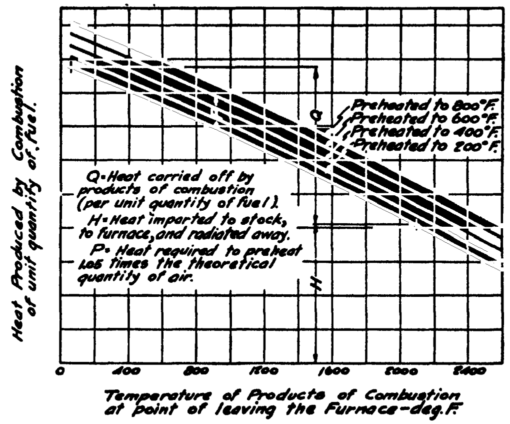


Fig. 141. Illustration of method of computing saving of fuel due to preheating of air.

The fuel saving that is indicated by equation 22 was calculated for rich fuels, such as natural gas, oil, or coal (see Fig. 142), and for lean fuels, such as raw producer gas (see Fig. 143), on the basis of the values given in Figs. 110, 111, and 116. At first glance it may appear surprising that the relative saving for a given temperature of preheat is greater when the flue-gas temperature is high than when it is low. However, the reason for this fact is very evident from the diagram, Fig. 141.

A similar reasoning may be followed if both the gas and the air are preheated; but, since the method of preheating both the gas and the air is now seldom followed in heating- or annealing-furnace practice, no calculations are offered here for that case. An additional effect, which

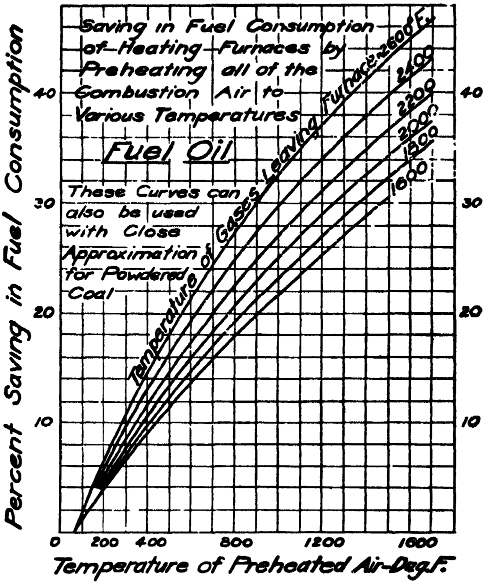


Fig. 142. Saving of fuel oil (or coal) caused by preheating of air.

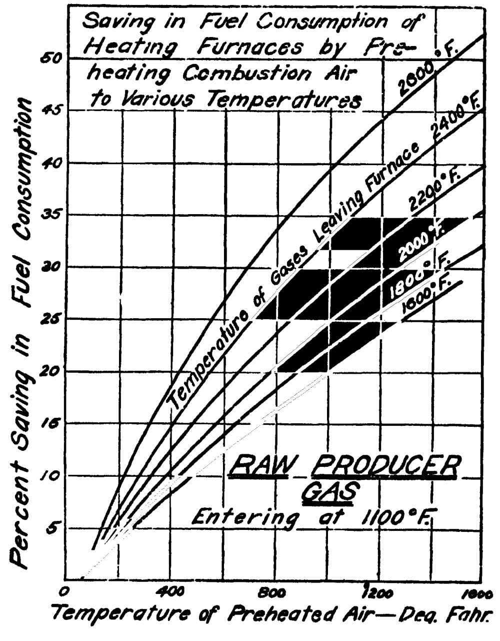


Fig. 143. Saving of producer gas caused by preheating of air.

is very important but rather intangible, is not shown in Figs. 142 and 143. It is caused by the preheating of combustion air and consists in quicker combustion.

With burners of usual design some excess air is needed to complete the process of combustion in the furnace, if the air is cold. If the air is hot, very little, if any, excess air is needed. A builder of recuperators, for that reason, increased by 5 per cent the savings that are read from Figs. 142 and 143. Preheated air shortens the flame and causes the so-called flame to become invisible if the air temperature is high enough. The quick combustion concentrates the heat near the burner and does not produce that long flame which extends all the way across the hearth. This may be corrected by working with a higher velocity in the burners. In either case the gases are clear, nonluminous unless the air temperature exceeds 1200 F. In that case, hydrocarbon fuels are partially cracked in the burner.

It should be noted that Figs. 142 and 143 refer only to furnaces having a constant temperature all over the hearth. They do not refer to those front-fired continuous furnaces that carry a drooping temperature.

Referring again to Figs. 142 and 143, it may be seen that very considerable fuel savings can be effected by preheating the combustion air, especially for furnaces in which high temperatures exist. The extent to which these figures can be realized and the difficulties in the path of such realization are mentioned in Chapter 7 and are discussed in Volume II.

(c) Recuperators.* In Chapter 1 it was shown that the two existing types of preheating devices are the recuperator and the regenerator. Proof is given later that, for all practical purposes, the calculations for a regenerator are identical with those for a counterflow tile recuperator. For that reason, recuperators are considered first.

Recuperators may be subdivided into three classes, according to the flow of the stack gases and of the air, although combinations of the classes are often found. The three types, counterflow, parallel-flow, and cross-flow, are diagrammatically shown in Figs. 144, 145, and 146. For the first two types, the distribution of temperature along the path of the gas and air flow can be represented graphically in a rather simple manner (see Fig. 147), but for the cross-flow, the diagram must, of necessity, be more complex. It is, in a measure, represented by

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* In this section, the term "recuperator" is applied to those heat exchangers in which the heat flows in a steady stream through a wall, it being understood (see page 12) that three or more regenerators used in rotation can be applied to recuperative furnaces.

Fig. 148. Only average temperatures (referring to up-and-down location) are shown, because the temperatures in and around the tubes vary from top to bottom.

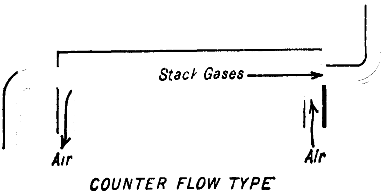


Fig. 144. Counterflow recuperator.

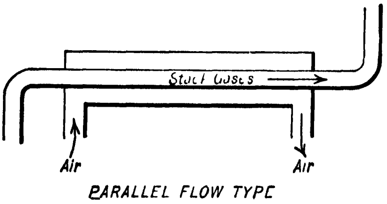


Fig. 145. Parallel-flow recuperator.

It may be readily observed from the illustrations that the counterflow type allows the highest temperature of preheat to be reached, whereas the parallel-flow type gives the lowest maximum temperature of the recuperator walls. For that reason it is sometimes used in connection with metallic recuperator walls, the temperature of which must be kept comparatively low. In some designs a combination of these types is adopted. The cold air is first caused to circulate around the hot end of the tubes or plates in order to keep them cool, and then is by-passed to the cold end, whence it flows with a combination of crossflow and counterflow toward the hot end again.

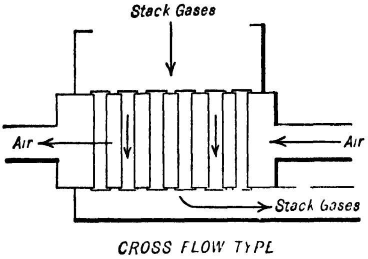


Fig. 146. Cross-flow recuperator.

If the coefficients of heat transmission were constant, the curves in Fig. 147 would be logarithmic. But it was shown in Chapter 2 that there is considerable variation in the value of the coefficient, depending on the temperature of gas and air, the density and velocity of gas and air, after-burning, radiation, leakage and the character of the heat-exchanging surface. Rigorous mathematical treatment of the problem of heat transmission in recuperators is, in consequence, out of the question, and approximate methods must be used. In view of additional variables entering into the problem, the necessity of approximation is no drawback, but is probably a blessing in disguise, because it keeps engineers from the erroneous belief that recuperators or regenerators can be calculated with precision.

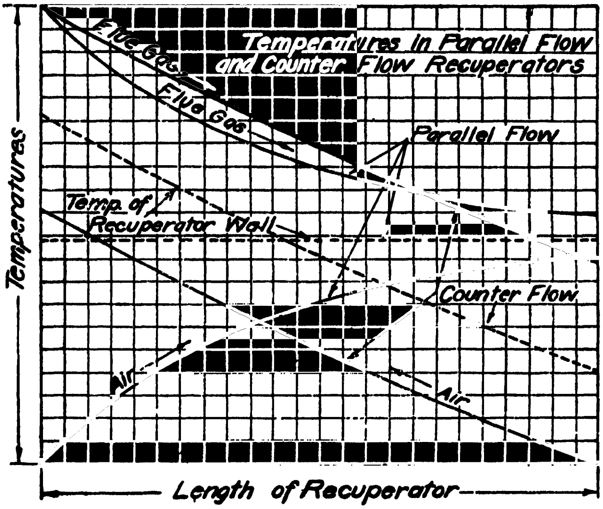


Fig. 147. Temperature distribution in recuperators.

The general equation for heat transmission in recuperators is: (weight of gas flowing in unit time, multiplied by temperature drop of flue gas, multiplied by average specific heat of flue gas) equals

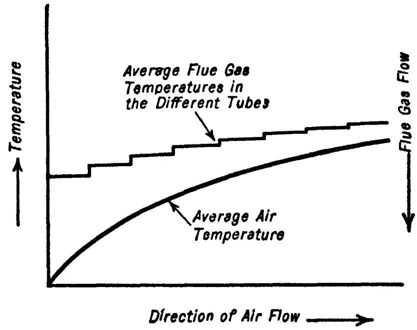


Fig. 148. Temperature distribution in a cross-flow recuperator.

(heating surface, multiplied by average heat-transmission coefficient, multiplied by mean temperature difference between flue gas and air) equals (weight of air flowing in unit time, multiplied by temperature rise of air, multiplied by average specific heat of air); or, with the notation of Fig. 149:*

$$W_g \times (T_{1g} - T_{2g}) \times c_g = A \times k \times (T_g' - T_a')$$
$$= W_a \times (T_{2a} - T_{1a}) \times c_a \qquad (23)$$

Applied to existing recuperators, the equation furnishes values of the heat-transmission coefficient k. For new recuperators, it determines the necessary amount of heating surface.

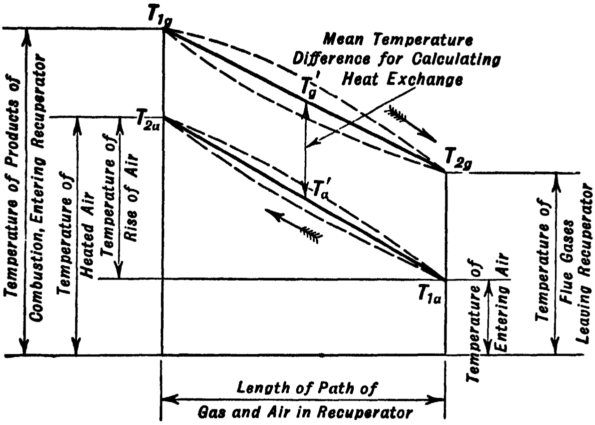

T_{1g}
Mean Temperature Difference for Calculating Heat Exchange
T_g'
T_{2a}
T_{2g}
T_a'
T_{1a}
Temperature of Products of Combustion, Entering Recuperator
Temperature of Heated Air
Temperature of Rise of Air
Temperature of Entering Air
Temperature of Flue Gases Leaving Recuperator
Length of Path of Gas and Air in Recuperator

Fig. 149. Temperature relations in a counterflow recuperator.

In either case, the recuperator must be free from leaks, if the results are to be trustworthy. The effect of leaks and of heat loss from the walls are discussed later.

The coefficient k, common units for which are Btu per (sq ft, hr, F), is the reciprocal of a resistance, which fact may be expressed by the equation $k = \dfrac{1}{R_g + R_w + R_a}$,

where R_g = resistance from flue gas to wall $= \dfrac{1}{k_g}$

R_w = resistance to heat flow through wall $= \dfrac{1}{k_w}$

R_a = resistance from wall to air $= \dfrac{1}{k_a}$

---

* In a cross-flow recuperator (Figs. 146 and 148), T_{2g} is the temperature of that portion of the flue gases which leaves the tubes in the center of the tube bank; T_{2a} is the temperature of the preheated air beyond the middle of the last tube. The heat-exchanging surface required by a cross-flow recuperator is greater than that required by a counterflow recuperator, for equal heat transfer. For a given heat-exchanging area, the heat transfers of parallel flow, cross-flow, and counterflow are roughly as 2 to 2.8 to 3, under average conditions.

The transfer of heat has already been discussed in Chapter 2 but needs amplification here. On the air side, heat transfer from the separating wall to the air takes place almost entirely by convection, the radiation-absorbing capacity of the small amount of water vapor present in the air being practically zero. The coefficient of heat transfer by convection increases rapidly with the mass velocity, i.e., the product of velocity times density of the air or gases, but is practically independent of the temperature. In Fig. 150 are given data on convective heat transfer, for flow along flat surfaces, through the inside of tubes, and across tube banks. For flat surfaces, the coefficient can be closely approximated by the straight-line equation

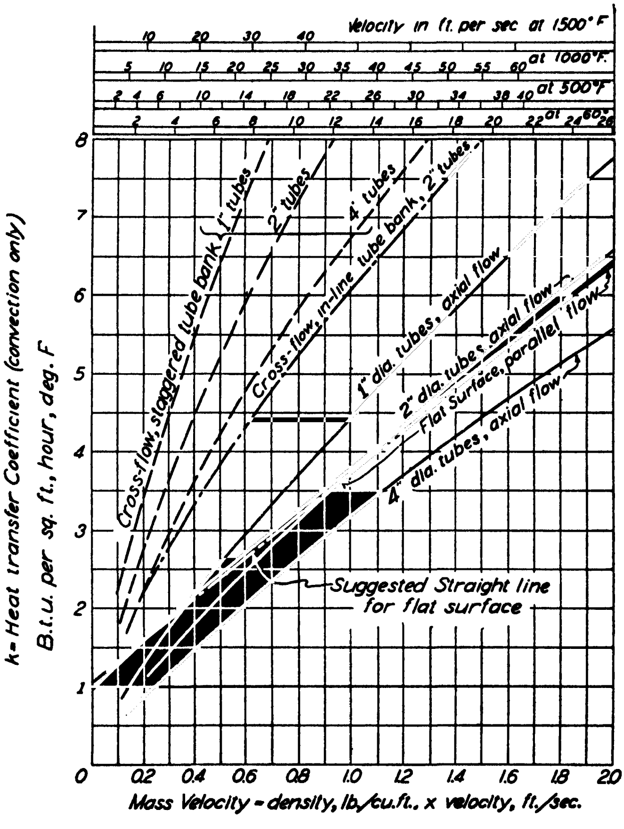


Fig. 150. Heat transfer by convection. Lower scale applies to all gases. Upper scales apply to air or to flue gases of same density as air.

$$k_a = 1.0 + 2.71qc \tag{24}$$

where k_a = coefficient of heat transfer from flat surface to air, Btu per (sq ft, hr, F); q = density of air, pounds per cubic foot; and c = velocity of flow, feet per second.

Fig. 150 also contains information on the heat-transfer coefficient from tube walls to air. The coefficient is proportional to the reciprocal of the fourth power of the tube diameter. This means that the heat-transfer coefficient in a 1-in. tube is $\sqrt[4]{4} = 1.4$ times as great as it is in a 4-in. tube. For each tube diameter a sufficiently close straight-line approximation can be devised.

For convective heat transfer from the products of combustion to the separating wall of the recuperator, practically the same relations hold. In addition, however, heat is transferred by gas radiation, which often far outweighs the effect of convection. As mentioned in Chapter 2, the coefficient of heat transfer by gas radiation is independent of the velocity of flow, but varies with the temperature of the gases, their composition, and the thickness of the gas layer. The values given in Fig. 43 are averages for the products of combustion, without excess air, of the rich fuels such as natural, coke-oven, or city gas, and oil or tar. They apply almost exactly to oil fuel; for the other rich fuels mentioned the radiation is 10 to 15 per cent greater, while for producer gas it is 5 to 10 per cent less, and for blast-furnace gas, 20 per cent less. The values must be multiplied by the radiation-absorption factor (emissivity) of the solid surface, usually 91 per cent for clean brick, tile, or rough iron surfaces and 0.70 for glazed brick surfaces.

For thicknesses of gas layer usual in recuperators and regenerators an increase (or decrease) of 1 per cent in the CO_2 content from the value of 12 per cent given in Fig. 43 raises (or lowers) the gas radiation about 1 per cent, whereas an increase (or decrease) of 1 per cent in the H_2O content raises (or lowers) the gas radiation about 1¾ per cent.

A few examples will illustrate the method of calculating the overall coefficient of heat transfer, which, as previously stated, has the form

$$k = \frac{1}{R_g + R_w + R_a} \tag{25}$$

EXAMPLE 1. Let flue gases having an average temperature of 1600 F flow in a passage 2 in. wide along one side of a flat recuperator wall with a velocity of 20 fps, while air at an average temperature of 300 F flows along the other side of the same wall at a velocity of 30 fps. What is the resulting heat-transfer coefficient?

If the wall is of metal, its resistance R_w is so small that it can be neglected.

The air at 300 F and 30 fps has the same mass velocity as air at 60 F moving with a speed of

$$\frac{460 + 60}{460 + 300} \times 30 = 20.5 \text{ fps}$$

Then, from Fig. 150, the value of k for flat surfaces is 5.2 Btu per (sq ft, hr, F).

The flue gases at 1600 F and 20 fps have a mass velocity the same as gases at 60 F moving at $\frac{460 + 60}{460 + 1600} \times 20 = 5.05$ fps; from Fig. 150, the corresponding convection coefficient is 2.12 Btu per (sq ft, hr, F). The gas radiation coefficient for 2-in. thickness of gas layer and 1600 F from Fig. 43, is 3.0, which must be multiplied by the absorption coefficient of 91 per cent for the rough metal wall, giving 2.73 Btu per (sq ft, hr, F).

$$k_g = 2.12 \text{ convection} + 2.73 \text{ gas radiation} = 4.85$$

Then,

$$k = \frac{1}{\frac{1}{5.2} + \frac{1}{4.85}} = 2.50 \text{ Btu per (sq ft, hr, F)}$$

for heat flow through a metallic wall.

EXAMPLE 2. In a tile recuperator, velocities are lower. In such a recuperator let the air velocity equal 10 fps, and let the air have an average temperature of 900 F; the flue gas travels at 6⅔ fps in passages of 6-in. width and has 1600 F average temperature. The recuperator wall of fireclay is 1 in. thick, having a conductivity of 0.75 Btu per (ft, hr, F); then,

$$k_a = 1.9 \quad \text{(from Fig. 150)}$$

$$k_g = 1.4 + (4.8 \times 0.91) = 5.7 \quad \text{(from Figs. 150 and 43)}$$

$$k = \frac{1}{\frac{1}{5.7} + \frac{1}{12 \times 0.75} + \frac{1}{1.9}} = 1.23 \text{ Btu per (sq ft, hr, F)}$$

As may be expected, the greatest resistance of the three contributing resistances largely determines the overall heat-transferring ability of the combination.

It is appropriate to mention at this point that at times the heat-transmission coefficients obtained from tests will be much higher than those obtained from Fig. 150 or from a calculation. Radiating solid matter in the products of combustion is responsible for the increase in heat transmission in some cases, but, more often, delayed combustion continuing in the recuperator causes an apparent rise in the heat-transmission coefficient. The rise is apparent only because heat which is generated in the recuperator by combustion causes the true temperature difference to be higher than the one used in the calculation. Indirect heating surfaces (not included in the calculation) also increase the apparent coefficient of heat transfer. Again, other tests indicate lower values of heat transmission than those indicated in Fig. 150. Several reasons may underlie an apparent reduction in heat transfer. There may be leakage between stack gas and cold-air ducts; in that case, the stack draft pulls partly preheated air into the stack, with the result that the stack gases are cooled and partly preheated air is lost. Again, there may be either infiltration of cold air into the furnace or discharge of hot gases from its door or cracks. In either case, the actual weight of flue gas differs from the weight theoretically required for combustion, and errors of calculation are likely to result.

On the air side, the heat-transfer coefficient grows with the velocity of flow of the air. It is therefore desirable, in order to reduce the size of the recuperator, to pass the air through at high velocities, wherever this is practicable (up to the speed at which the increase of cost of power for moving the air against the corresponding pressure, balances the reduction of cost of the recuperator). On the flue-gas side, however, this rule does not hold; an increase of velocity, although it increases the convective heat transfer, requires that the gas passages be reduced in width (for a given quantity of gases), and thereby decreases gaseous radiation. The net result may actually be a decrease of the total heat-transfer coefficient on the gas side; thus in the previous examples, tripling the gas velocity *reduces* k_g by 15 per cent, instead of increasing it.

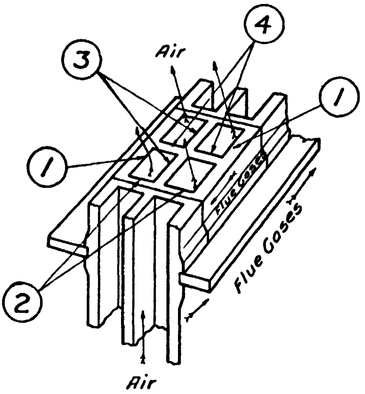


Fig. 151. Recuperator tile with indirect heating surface. (1) indicates direct heating surface, the other numbers indirect heating surface.

The best design of recuperator, therefore, from a heat-transfer standpoint, is that in which the flue gases travel slowly in large passages, while the air is passed through at high velocity. In metallic recuperators the latter requirement is easily fulfilled, but in tile recuperators the joints cannot be made and kept tight, in spite of the greatest care, and even a slight excess of pressure of the air over that of the gases causes excessive leakage. It is best, then, in tile recuperators, to keep both air and gas velocities low, and try to make up for the reduced convection heat transfer by providing secondary or indirect heating

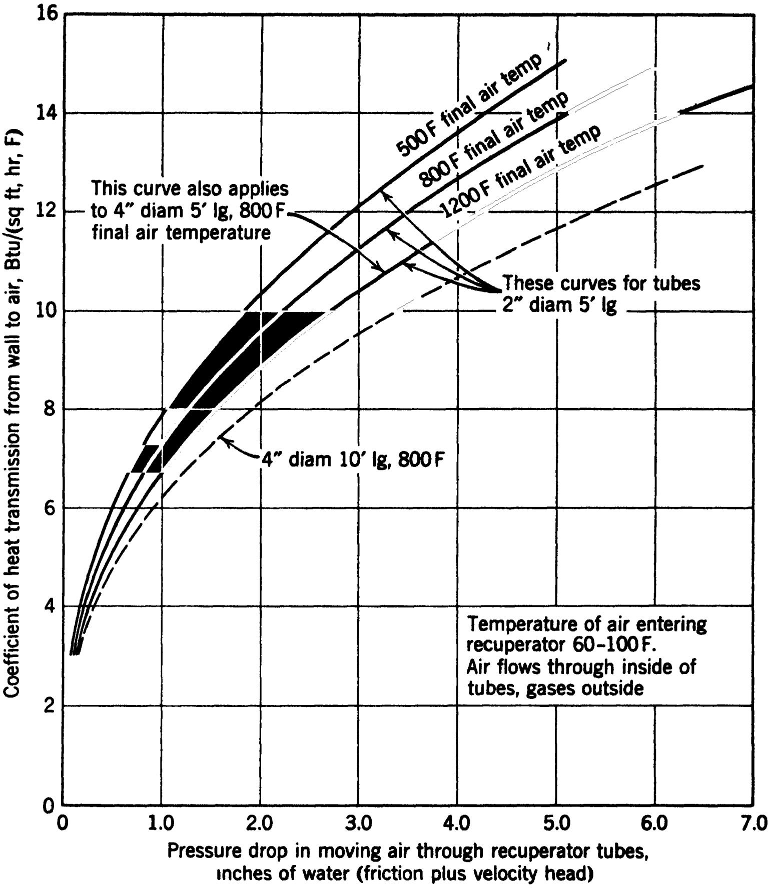


Fig. 152. Heat-transfer coefficient from recuperator tubes to air, as a function of pressure drop.

surface inside the air passages. An example is shown in Fig. 151. The surfaces of the partitions receive heat by radiation from the outer walls of the passage and transmit it to the air by convection. Calculation shows that at the hot end of the tile recuperator, 1 sq ft of this indirect heating surface is equivalent to 0.53 sq ft of direct heating surface, whereas at the cold end it is equivalent to 0.27 sq ft of direct heating surface. On the average, therefore, the ribs or partition walls increase the overall effectiveness of heat transfer from gases to air by about 40 per cent.

For the purpose of furnishing a general conception of the relation between pressure drop and heat-transfer coefficient, Fig. 152 is offered. Variations in recuperator design do not permit this relation to be in-variable for all recuperators, and deviations from these values are found in practice; but they are sufficiently accurate for estimating the heat-transfer coefficient that can be expected for a given pressure drop of the air in passage through the recuperator.

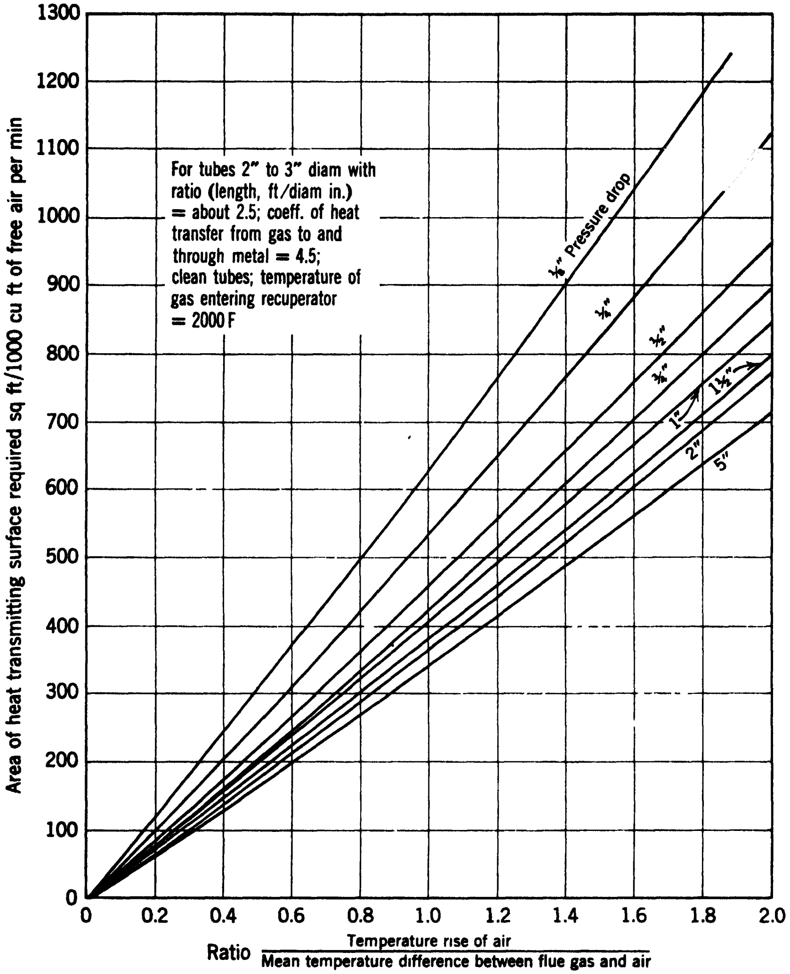


Fig. 153. Heating surface required in metallic recuperators, as related to pressure drop in inches of water.

Based on the values of Fig. 152, the curves of Fig. 153 show the amount of heating surface required for a recuperator, as related to the available pressure drop of the air and the desired temperature rise of the latter. The curves are based on an average value of 4.5 for the heat-transfer coefficient on the gas side. This cannot, of course, be correct for all conditions, but it is a fairly representative value.

The use of Fig. 153 will be illustrated by an example:

EXAMPLE. The combustion air for a furnace burning 20 gal of oil per hr is to be heated from 60 to 500 F. Furnace temperature 2200 F. A pressure drop of 1 in. of water can be allowed in the recuperator. What area of heat-transmitting surface is required in the (metallic) recuperator?

The amount of air required is 14.068 lb per lb of oil 5 or 95 lb per gal, or

$$\frac{95 \times 20 \times 13.08 \text{ cu ft per lb air}}{60} = 414 \text{ cfm}$$

The gases will probably enter the recuperator at 2000 F. They drop about 7/10 degree for each degree rise of the air; hence the gas leaving temperature is

$$2000 - \tfrac{7}{10}(500 - 60) = 1692 \text{ F}$$

Mean temperature difference between gases and air

$$= \frac{2000 - 500 + 1692 - 60}{2} = 1566 \text{ F}$$

$$\text{Ratio } \frac{\text{Temperature rise}}{\text{Mean temperature difference}} = \frac{500 - 60}{1566} = 0.28$$

The required heating surface from Fig. 153 is 110 sq ft per 1000 cu ft of free air per min, or

$$\frac{414 \text{ cfm}}{1000} \times 110 = 46 \text{ sq ft}$$

From Fig. 153 it may be seen that, as the pressure drop is increased more and more, the recuperator size is reduced more and more slowly.

At first thought it might appear that it does not pay to increase the pressure drop in the air passages beyond 2 in. of water column. Actually, metallic recuperators are designed for a pressure drop of 5 to 8 in. of water column, for the purposes of reducing the first cost and of protecting the heat-transmitting wall.

The following facts must also be considered: The heating rate in the furnace may at times have to be increased beyond the normal rate; for this purpose the fuel and air flow have to be boosted. The pressure drop grows as the square of the air flow; and, since fans, which usually supply the air, are sharply limited in the pressure they can generate, the desired increase in air flow cannot materialize, unless the normal pressure drop through the recuperator is much smaller than the pressure which the fan can generate.

The actual calculation of that recuperator heating surface that is required to heat the combustion air for a given furnace to a given temperature is more or less tedious, even if the relation of Fig. 153 is used. For that reason, three graphs, for counterflow metallic recuperators, are offered in Figs. 154 to 156, by which such calculations are rendered unnecessary, except for checking purposes.

The temperature relations underlying the calculation of Figs. 154 to 156 are illustrated in Fig. 149, already referred to. The general equation for the calculation of recuperators had already been given on page 214 as equation 23. On the basis of straight-line relations, the active temperature difference equals

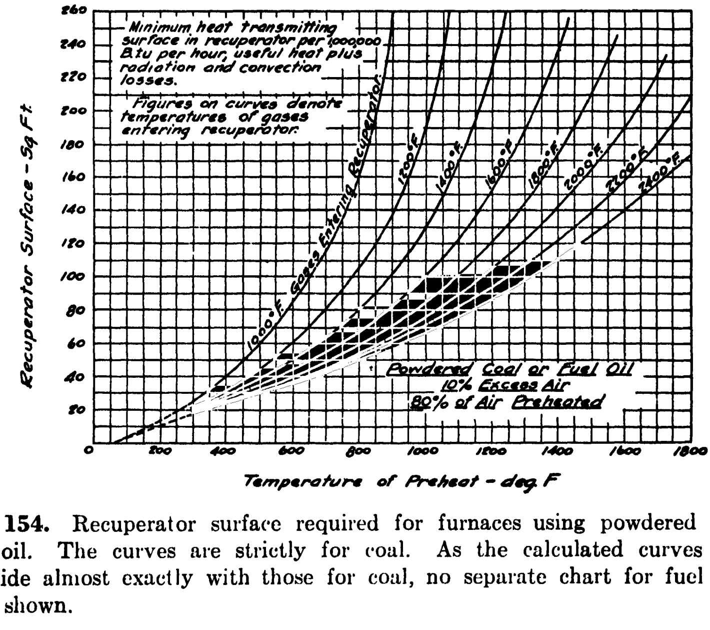


Fig. 154. Recuperator surface required for furnaces using powdered coal or fuel oil. The curves are strictly for coal. As the calculated curves for oil coincide almost exactly with those for coal, no separate chart for fuel oil has been shown.

$$T_g' - T_a' = \tfrac{1}{2}(T_{1g} + T_{2g}) - \tfrac{1}{2}(T_{1a} + T_{2a}) \qquad (26)$$

As a rule, that heating surface A is to be determined that is required to heat a given weight of air per hour (W_a), having a mean specific heat c_{pa}, from an initial temperature T_{1a} to a final temperature T_{2a}. Then,

$$W_a c_{pa}(T_{2a} - T_{1a}) = W_g c_{pg}(T_{1g} - T_{2g})$$

from which

$$T_{1g} - T_{2g} = (T_{2a} - T_{1a})\frac{W_a c_{pa}}{W_g c_{pg}}$$

and finally

$$Q = W_a c_{pa}(T_{2a} - T_{1a}) = kA\,\frac{1}{2}\left[2T_{1g} - T_{1a} - T_{2a} - (T_{2a} - T_{1a})\frac{W_a c_{pa}}{W_g c_{pg}}\right] \qquad 27)$$

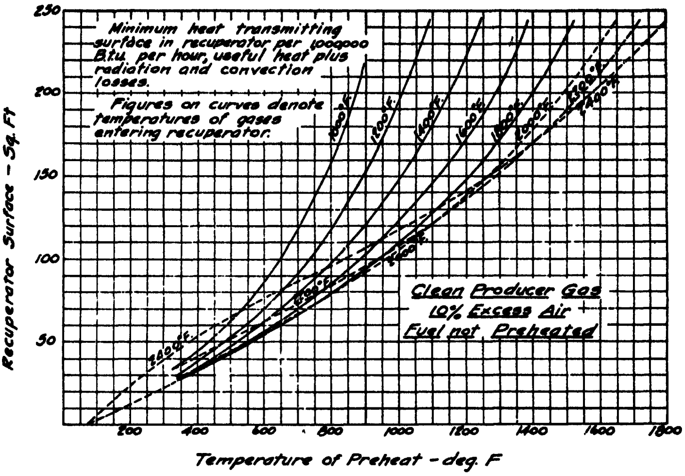


Fig. 155. Recuperator surface for furnaces using producer gas.

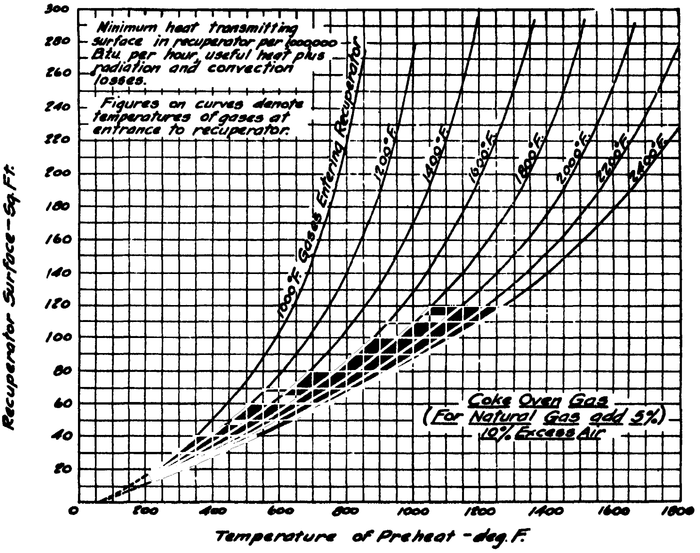


Fig. 156. Recuperator surface for furnaces using coke-oven gas. This chart can be used also for natural gas, by adding 5 per cent to the areas.

This equation is easily solved for the heating surface A. The solution for the final air temperature T_{2a} is more cumbersome. Instead of hourly weights of flue gas and air, in connection with weight-specific heats, hourly volumes, referred to standard conditions, can just as well be used, provided that volume-specific heats are used.

The supposedly more correct logarithmic equation is

$$Q = kA \frac{(T_{1g} - T_{2a}) - (T_{2g} - T_{1a})}{\ln\left(\dfrac{T_{1g} - T_{2a}}{T_{2g} - T_{1a}}\right)} \tag{28}$$

In order to solve for A, we must eliminate T_{2g}. This is done, as previously, from

$$T_{1g} - T_{2g} = (T_{2a} - T_{1a}) \frac{W_a c_{pa}}{W_g c_{pg}}$$

The algebraic substitution is cumbersome. In each individual case it is easier to compute T_{2g} from the last equation and to substitute its numerical value in the logarithmic equation.

For computing the recuperator surface, the heat to be transferred through that surface must be known. It is determined as follows: For a given furnace, the hourly fuel consumption is determined as was shown in Chapter 5. However, preheated air is used in the calculation, whereby the fuel consumption is correspondingly reduced. The hourly flow of air and of flue gas is thus established. Multiplying hourly air flow by desired temperature rise of air and by mean specific heat of air furnishes the heat to be transferred through the recuperator walls each hour.

Figures 154 to 156 are of interest to the furnace engineer who, from his calculations knows the quantity of heat that is to be left in the furnace per hour (heat to charge, wall losses, heat to water-cooled parts, etc.). But the builder of recuperators is interested in calculating the recuperator surface from the cubic feet of air that are to be heated per minute to a given temperature. This latter figure can be derived from Figs. 154 to 156 in the following manner: 1,000,000 Btu (= heat left in furnace) = combustion heat + heat in air — heat in waste gases. In developing the relation, the premises are the same as before, namely 10 per cent excess air, and 80 per cent of the combustion air is preheated. Since the calculations are approximations, the specific heat of gases is taken to be 0.25.

For the purpose of this calculation, let

N = number of fuel units per hour
L = lower heating value of fuel per unit
W_a = pounds of air per unit of fuel for perfect combustion
W_g = pounds of flue gas per unit of fuel with perfect combustion

$$\begin{aligned} NL + 0.8 \times 1.1 \times 0.25W_a(T_a - 60) \\ = 1{,}000{,}000 + 0.25N(0.1W_a + W_g)(T_g - 60) \end{aligned}$$

$$N = \frac{1{,}000{,}000}{L + 0.22W_a(T_a - 60) - 0.25(0.1W_a + W_g)(T_g - 60)}$$

Volume of air heated per minute

$$\text{cfm} = \frac{0.8 \times 1.1W_a \times 13 \times N}{60}$$

Substitution for N furnishes B, cfm of air preheated for 1,000,000 Btu per hr left in furnace

$$B = \frac{192{,}500W_a}{L + 0.22W_a(T_a - 60) - 0.25(0.1W_a + W_g)(T_g - 60)}$$

If A is the recuperator surface per 1,000,000 Btu read from Figs. 154 to 156 and if B cfm are needed per 1,000,000 Btu, then the recuperator surface needed to heat 1000 cfm is $1000A/B$, which equals

$$\text{Multiplier} = \frac{L + 0.22W_a(T_a - 60) - 0.25(0.1W_a + W_g)(T_g - 60)}{192.5W_a}$$

EXAMPLE. From Fig. 154 (fuel oil) select $T_a = 800$ F and $T_g = 2000$ F, then the multiplier equals

$$\frac{17{,}340 + 0.22 \times 13.57 \times 740 - 0.25(1.36 + 14.57)1940}{192.5 \times 13.57}$$

which equals 4.56. Then recuperator area (sq ft per 1000 cfm) equals 55 (from Fig. 154) × 4.56 = 250 sq ft.

It may again be mentioned that Figs. 154 to 156 were plotted for an overall coefficient of heat transfer equal to 2.5. If the coefficient equals 5.00, only half of the indicated area is needed.

As previously stated, straight-line relations were used to obtain equation 26 and for computing the graphs of Figs. 154 to 156. Checking calculations were made to ascertain how the results would vary, if the logarithmic relation had been adopted for calculation. The difference was found to be negligible except in that region where T_{2a} approaches T_{1g} very closely. In that region, recuperators become impracticably large and expensive; this means that for all practical purposes, the straight-line relation gives sufficiently accurate results.

The heating surface represented in the diagrams (Figs. 154 to 156) was calculated on the basis of an overall heat-transmission coefficient of 2.5 Btu per (sq ft, hr, F). This value is easily attained in metallic recuperators. At this point it may be well to discuss the heat-transfer coefficients actually obtained in recuperator tests. Makers of recuperators sometimes claim to obtain coefficients of 5 to 10. Although it is true that, owing to the causes mentioned on page 217, apparent values higher than 2.5 are sometimes obtained, it is unsafe to design recuperators of conventional types on the basis of these figures. In very carefully conducted tests of recuperators, in which the air and gas were metered and the gas temperatures were measured with aspiration pyrometers, the highest coefficients found by the authors were: for a cast unit recuperator, surrounded by brick walls which acted as indirect heating surface (not included in the area on which the coefficient was based), and exposed to some direct radiation, 4.7; for a tubular recuperator, with the air passing through at high velocity, 5.3; and for a silicon carbide recuperator with "corebusters" (indirect, not-included heating surface), 3.2 Btu per (sq ft, hr, F). The only recorded case in which a coefficient much higher than 5 was obtained in a careful test, was that of a recuperator in which the tubes were fully exposed to direct radiation from the interior of a high-temperature furnace. Such a recuperator, however, cannot be classed as a heat-saving device, because it takes useful, high-potential heat directly from the heating chamber, instead of extracting waste heat from the flue gases.

On the other hand, the coefficient is often found to be lower than 2.5. In addition to the reasons mentioned on page 217, there is the effect of losses from the walls, and of deposits of dust and soot on the surfaces, as well as glazing or slagging in the case of tile recuperators. With oil fuel, instances have been known where the tubes became practically choked with soot after several months' operation. Some oils contain a small amount of saltwater (calcium sulphate), .which is atomized with the oil and is evaporated. The salt deposits itself on the recuperator surfaces and reduces the rate of heat transfer.

Summing up, it may be stated that the heat-transfer coefficients in recuperators lie within the following ranges:

Type of recuperator	Btu per (sq ft, hr, F)
Clay tiles	0.4 to 1.7
Silicon carbide with vortex corebusters	3 to 5
Metallic, with low air velocity	2 to 3.5
Metallic, with high air velocity	3.5 to 5.5
Metallic, with high air velocity and with projections or fins which are not included in (calculation) heating surface	7 to 10

The curves can be used with reasonable accuracy for temperatures of entering air other than 60 F, by reading from values of preheat and of furnace temperature diminished by the difference between entering air temperature and 60 F. For instance, if the furnace temperature equals 2000 F, and preheat 1000 F, and air enters the recuperator at

160 F, read heating-surface requirement from the chart by interpolating for 1000 − (160 − 60) = 900 F preheat and 1900 F furnace temperature.

The areas of the recuperator surfaces shown by the charts are minimum values unless higher coefficients of heat transfer are obtained. In practice, larger surfaces must be provided for several reasons: part of the flue gases escape around the furnace doors; heat is lost by radiation from the outer surface of the recuperator; combustion is not perfect; there is leakage and drop of temperature of preheated air between recuperator and burner; true countercurrent conditions do not exist. **Twenty-five per cent excess area will provide a good margin of safety.**

The heating surfaces represented in the diagrams (Figs. 154 to 156) apply to furnaces requiring 1,000,000 Btu per hr. This figure represents the difference between the sensible heat of the fuel mixture plus combustion heat, and the sensible heat in the products of combustion (at the point of leaving the furnace) per hour. It also represents that hourly heat quantity that is necessary to heat the stock and the furnace walls, and to cover heat losses of all sorts, such as losses to the walls, to cooling water, and radiation through openings. If the recuperator is to be used for a furnace requiring H Btu per hr, then the heating surfaces obtained from the curves must be multiplied by $H/1{,}000{,}000$.

The following examples will illustrate the use of the charts:

Example 1. An oilfired furnace operating at 2300 F and burning 30 gal of oil 5 per hr is to be equipped with a recuperator to preheat 80 per cent of the combustion air to 600 F. The flue gases contain 2 per cent CO and 2 per cent H_2. How large an area of recuperator surface is required, if the gases reach the recuperator at 2200 F?

From Fig. 142, the reduction of fuel consumption to be expected is 18 per cent. The heat developed by combustion is 18,830 Btu per lb of oil (lower heating value). The heat taken in by the preheated air is 80% × 1700 = 1360 Btu per lb oil. The sensible heat in the waste gases = 9550 Btu, and the loss by incomplete combustion (no excess air) is 2040 Btu. The heat left in the furnace is 18,830 + 1360 − 9550 − 2040 = 8600 Btu per lb of oil, or 8600 × 6.75 lb per gal × 30 × 82% = 1,430,000 Btu per hr. From Fig. 154, the heating surface required is 37 sq ft per million Btu per hr left in the furnace (useful heat plus wall losses).

$$\frac{1{,}430{,}000}{1{,}000{,}000} \times 37 = 53 \text{ sq ft}$$

Allowing 25 per cent as a margin gives 66 sq ft as the surface required in the recuperator. (After-burning may, in some cases, allow the same preheat to be obtained with a considerably smaller surface, but metallic recuperators subjected to after-burning usually last only a short time.)

EXAMPLE 2. A tile recuperator of 600 sq ft area, connected to a coke-oven gasfired furnace using 2,500,000 Btu per hr (useful heat plus wall losses) preheats the combustion air to 1400 F, the gases entering the recuperator at 2200 F. A duplicate furnace is planned, and it is proposed to install a recuperator of the same type but of 800 sq ft area. What preheat temperature may be expected from this larger recuperator?

From Fig. 156, the required heating surface is 152 sq ft per million Btu, or 380 sq ft total, based on a coefficient of 2.5. Hence the effective coefficient must be 380/600 × 2.5 = 1.59 Btu per (sq ft, hr, F). If this same coefficient applies to the large recuperator also, then the equivalent area for 2.5 Btu (corresponding to Fig. 156) would be 1.59/2.50 × 800 = 508 sq ft, or 508 × 1,000,000/2,500,000 = 203 sq ft per million Btu. By interpolating in Fig. 156 to find the intersection of the 203 sq ft line with the 2200 F furnace temperature curve, the preheat is found to be exactly 1600 F, or 200 F more than in the smaller recuperator.

In this example, the reduction of the quantity of air to be heated, due to the decrease of fuel consumption corresponding to the additional preheat, has not been taken into account, but the error caused thereby is negligible.

Ceramic as well as metallic recuperators have advantages and disadvantages. In ceramic recuperators, very high air temperatures can be attained, but the probability of leakage prevents delivery of combustion air to the burners at any desirable pressure. And leakage is harmful, as explained below. Before 1940, many failures of metallic recuperators were reported whenever it was attempted to obtain high air temperatures in connection with furnaces that carried temperatures of 2200 F or higher. Improvement in the design and the use of more suitable metallic materials have practically eliminated failures.

With ceramic recuperators, it is difficult to maintain the optimum fuel-to-air ratio, because leakage varies with time. Frequent analysis of the heated combustion air and of the furnace gases is helpful in discovering leaks and in maintaining the correct fuel-to-air ratio.

Design features and economies of recuperation with intermittent operated furnaces are discussed in Volume II.

For the sake of completeness, the term "pickup" which has crept into recuperator language must be explained. This term, which is usually expressed in per cent, means the actual rise of air temperature to the maximum possible rise of air temperature. With the notations of Fig. 149, the pick-up is $(T_{2a} - T_{1a})/(T_{1g} - T_{1a})$. Builders of recuperators limit themselves to a pickup ratio of 60 per cent. They go up to 75 per cent whenever fuel is very expensive. Figures 154 to 156 teach that high pickup ratios result in very large and expensive recuperators.

Some of the difficulties that are inherent to recuperators are avoided by regenerators, the theory of which will now be developed.

(d) Regenerators. A general view of a regenerator is given in Fig. 19. A clearer view is shown in Fig. 157.

Regenerators were originally invented, not for the purpose of heat salvage, but for the purpose of raising flame temperatures to high levels, so that industrial processes requiring very high temperatures could be carried out. Regenerative furnaces, for reasons explained in Volume II, have a rather limited application in the field of industrial heating of metals. If, in spite of this fact, a great deal of space is devoted to regenerators in the present volume, it is because there exists, at this writing, no easily understandable treatise on regenerators in the

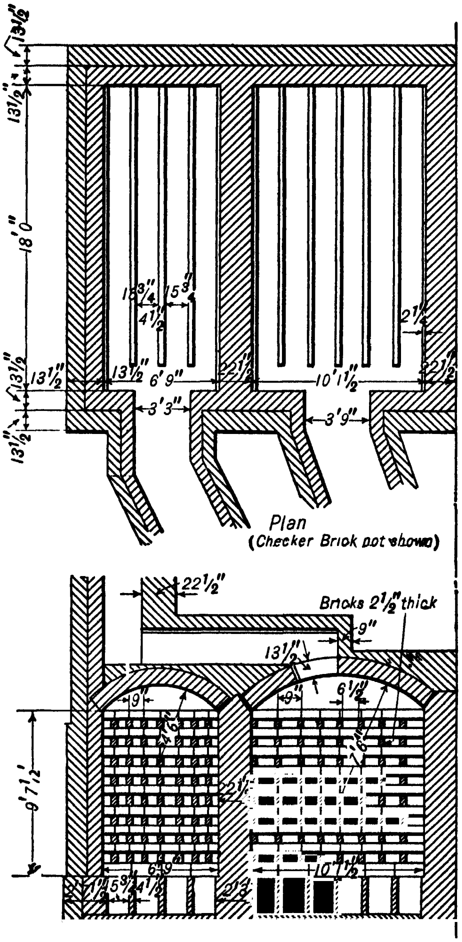


Fig. 157. Heat regenerator.

English language. The former editions of this book were used, not only for the design of industrial (i.e., metal-heating) furnaces, but also in connection with open-hearth furnaces, glass tanks, coke ovens, zinc-smelting furnaces, and the like. Those who have used previous editions and now refer to the present book will, therefore, expect that the section on regenerators has been brought up to date, even if the theory takes a little more space than would be warranted by the application of regenerators to industrial furnaces alone.

Any reasonably complete theory of regenerators (such as, for instance, developed by A. Schack in "Der Industrielle Wärmeübergang," fifth edition, Verlag Stahleisen) is too complicated to be relished by the average furnace engineer. A complete theory has to take care of a very large number of variables, among which are the following:

1. Temperature of products of combustion at entrance to regenerator.
2. Temperature of air leaving regenerator (variable with time).
3. Time on flue gas.
4. Time on air.
5. Thickness of checkerbricks.
6. Thermal conductivity of checkerbricks.
7. Specific heat of checkerbricks.
8. Arrangement of checkerbricks.
9. Weight of checkerbricks.
10. Extent of heat-exchanging surfaces.
11. Velocity of flow of flue gas.
12. Velocity of flow of air.
13. Dust or flame content of products of combustion.
14. Heat-transmission coefficient from flue gas to bricks.
15. Heat-transmission coefficient from bricks to air.
16. Distribution of gas or air over cross section of regenerator.
17. Heat transmitted in unit time.
18. Heat loss through enclosure.
19. Leakage (in or out) through enclosure.
20. Draft loss through regenerator.

For the purpose of the present discussion, the problem of taking care of so many variables is solved by investigating the influence of some of the variables, by setting practical limits for them, and by deriving relations between the remaining variables, with average constant values in the place of the other variables.

Important relations exist between size of furnace (and regenerator), time between reversals, thickness of brick, conductivity of brick, and heat-storage ratio of brick. In order to understand these relations, it is advisable to follow the events that occur during reversal. At the time of reversal, the products of combustion in the flue, regenerator, and uptake (indicated by shading in Fig. 158) must pass the burner before preheated air is ready to sustain the process of combustion in front of the burner. The time required for this process is the sum of the time elements required for throwing the valve, bringing the moving column of products of combustion to rest, starting them in the opposite direction, and passing them up to the burner. Doubtless, some mixing of waste products and incoming air occurs, so that the arrival of pure air in the furnace is still further delayed. The whole process consumes 5 to 30 sec, depending on the size of the regenerator. During part of the time a mixture of waste gases and air sweeps over the hearth of the furnace, producing oxidation. In furnaces which are fired with producer gas, and in which both air and gas are regenerated, an additional delay is due to the escape of at least a regenerator full of gas to the stack at each reversal. All these facts favor a long period of reversal; moreover, too frequent reversals distract the attention of the heater from other, equally important work, unless the reversing is done automatically by a time clock.

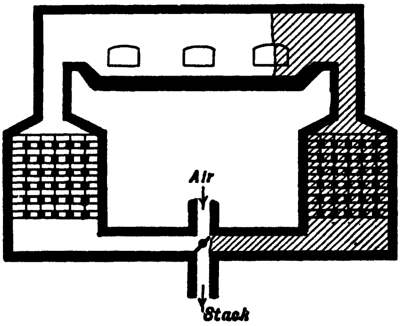


Fig. 158. Diagram illustrating events at time of reversal.

On the other hand, long periods between reversals necessitate a great weight of brickwork in the checker, because of the great quantity of heat that must be stored up between reversals. This results in larger regenerators and in greater first cost. Furthermore, long periods of reversal result in lower average temperature of preheat, and consequently in reduced fuel economy.

Since there are arguments in favor of both short and long periods of reversal, it follows that there will be a most favorable period for each furnace, and that a considerable departure, on either side, from the most favorable period will produce very little change in fuel economy.

The result is that a great variety of reversal periods are found in practice; they range from 5 min (in small semi-portable furnaces) to 15, or even 20 min (for large, four-door furnaces, such as are used in rolling mills). In blast-furnace stoves, the time between reversals is much longer. Moreover the gas period and the air period are different.

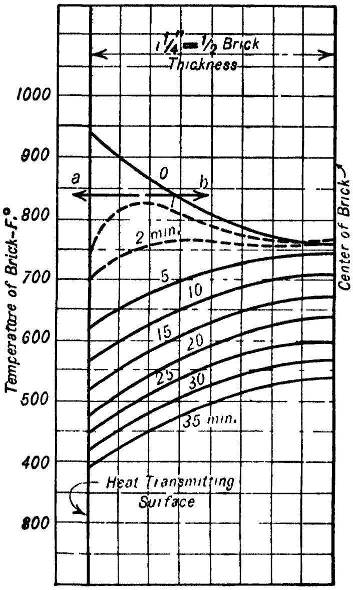


Fig. 159. Test data on temperature in a regenerator brick as a function of time and of place.

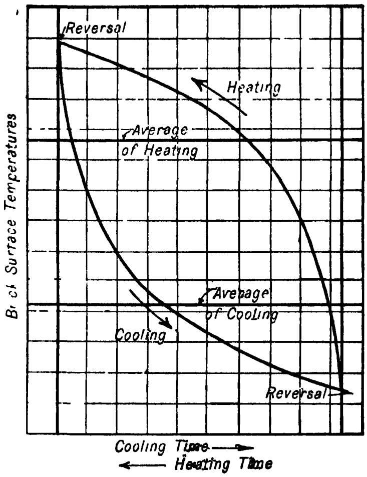
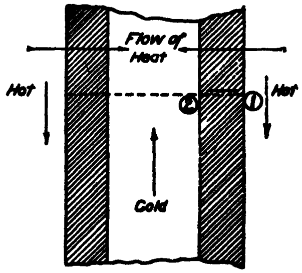


Fig. 160. Temperature cycle of surface of regenerator brick.

For a thorough understanding of the occurrences in a regenerator, a study of Figs. 159, 160, and 161 is very helpful. Figure 159, which was plotted from test data, shows how the temperature in a regenerator varies, for a brick of 2½-in. thickness, with time and with space, immediately after the reversal from heating to cooling. The figures on each line give the minutes that have elapsed between the time of the reversal and the time for which the temperature distribution is indicated. Immediately after the reversal, heat flows both ways (see arrows *a* and *b*), but the skin temperature drops so fast that heat flow into the brick ceases after 1 min, and the almost parabolic lines, marked 5, 10, 15, 20 min, etc., establish themselves. The skin temperature change in the heat-exchanging skin is most marked in the first minute after reversal, from which the conclusion follows that the time average of temperature of regenerator surface will, during the period of heating, be much greater than during the period of cooling. The same fact is most forcefully expressed in Fig. 160, in which the temperature cycle of the surface of the brick is expressed in a general way. By the latter statement is meant that the temperature rise and the time of cycle may vary within considerable limits, but that the temperature-time diagram will always show one characteristic, namely a very quick change of temperature immediately after the reversal and a gradual slowing down of the rate of temperature change. The effect is that heat flows from the products of combustion into the brick through a surface, which, on the average, is hotter than that through which the heat flows from the brick to the air that is being preheated. But that is exactly what happens in a thick-walled recuperator (see Fig. 161), in which point 1 has a temperature considerably in excess of that of point 2. With that knowledge, the problem of determining the correct size of a regenerator is reduced to the problem of calculating a recuperator.

Fig. 161. Thick-walled recuperator.



In this connection, two items need further attention. First, proof must be given that the regenerator is equivalent to a recuperator all the way through, and not only in spots. Second, the correct heat-transmission coefficient must be determined.

With reference to the first of these two points, the essential features with reference to temperature distribution are diagrammatically shown in Fig. 162. In this diagram, the abscissae represent position in regenerator with regard to path of flue gases and of air, while the ordinates represent temperature. The solid lines represent temperatures for a certain time between reversals; the dotted lines hold for a shorter time between reversals. While the temperature range between the final values varies considerably with the period between reversals, the average values vary less, so that, to avoid confusion, only one line was used for them in the diagram. The temperature lines were made straight, although, in reality, there is usually a slight curvature. If the heat-transmission coefficients were constant, the average temperature lines would be logarithmic; but the coefficient is far from being constant, as will presently appear. Evidently, the lines of average temperatures show the same characteristics as those of recuperators.

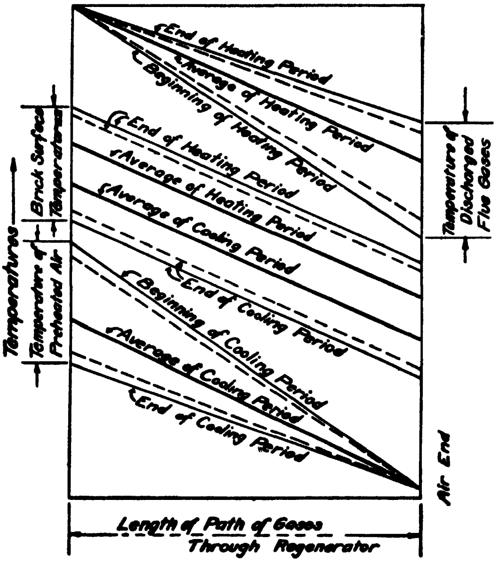

End of Heating Period
Average of Heating Period
Beginning of Heating Period
End of Heating Period
Average of Heating Period
Average of Cooling Period
End of Cooling Period
Beginning of Cooling Period
Average of Cooling Period
End of Cooling Period
Temperatures
Temperature of Preheated Air
Brick Surface Temperatures
Temperature of Discharged Flue Gases
Air End
Length of Path of Gases Through Regenerator

Fig. 162. Temperature distribution in a regenerator.

The second point, namely, the determination of the correct heat-transmission coefficient, needs further discussion. The overall heat-transmission coefficient,* just as in the case of recuperators, is

$$k = \frac{1}{1/k_g + 1/k_w + 1/k_a} \tag{29}$$

where k_g is coefficient of heat transfer from waste gases to brick surface and k_a is the coefficient from brick surface to the air, but in this case k_w is a coefficient of heat transmission into and out of the bricks. This brick coefficient k_w takes the place of the term (conductivity ÷ thickness) which appears in the formula for a thick-walled recuperator. In the determination of its value, brick conductivity, specific heat, brick thickness, and period between reversals play important parts. Furthermore, this coefficient varies widely with the nature of the temperature-time relations for the gases and the air: when the latter follow a loop curve as in Fig. 163*b*, the coefficient is more than twice as great as when they remain constant throughout the period, as in Fig. 163*a*. This was determined by working out the temperature-time curves for the brick surface by means of the Schmidt graphical method, also taking account of the great difference between the values of k_g and k_a, which are discussed below. In an actual regenerator, near the top the gas temperature is nearly constant while the air temperature follows the lower part of the "loop" curve. Near the bottom the air temperature is constant while the gas temperature follows the upper part of the "loop" curve. In the middle of the height both gas and air follow that form of curve. It might be thought, therefore, that coefficients derived on the basis of gas and air temperatures varying with time according to a curve of that form would more nearly represent actual conditions, but on the contrary the tests of Schumacher* on an actual experimental regenerator gave coefficients which check quite closely with those obtained by the graphical method on the basis of constant air and gas temperatures. The latter have, therefore, been used in Fig. 164, which shows how the equivalent brick coefficient varies with period of reversals and with brick thickness, for ordinary fireclay bricks. The experimental results of Schumacher have also been shown; his tests, however, were all made with only one brick thickness, namely, 3.2 in.

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* In the present book, the units for the coefficients of heat transfer are Btu per (sq ft, hr, F). In German treatises on regenerators, the heat-transfer coefficient is expressed in Btu per (sq ft, F, period), or rather in the equivalent metric dimensions Cal. per (sq m, C, period). If the reversal period is 1 hr the coefficient per hour and the coefficient per period are equal. The German method is of advantage in those cases where the heating and cooling periods are different, as, for instance, in blast-furnace stoves.

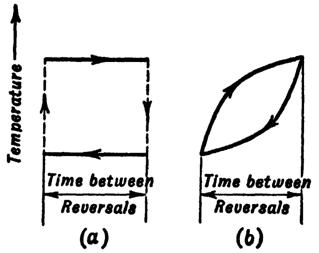

Fig. 163. Temperature variations of air and gases at a given place in a regenerator.

It should be remarked here, first, that the coefficient k_w is also affected by changes in the gas and air coefficients k_g and k_a, but within the range of variation of these usually encountered in regenerators the change in k_w is negligible; second, that all the coefficients are based on a heat transmitting area equal to the exposed heat-exchanging surface of the checkerwork on **one side** of the furnace **only.** The reason for taking the area of one side instead of that of both sides is that at all times heat is flowing *in* through the brick surface on one side of the furnace while at the same time it is flowing *out* through the surface on the other side. Hence, the area corresponding to the mean area in a recuperator is the regenerator surface of one side and not the area of both sides.

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* *Mitteilungen 141 der Wärmestelle des Vereins deutscher Eisenhüttenleute,* August 1930.

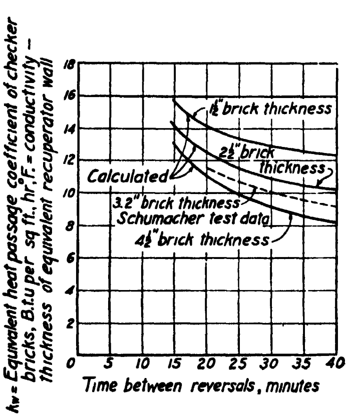

Fig. 164. Equivalent heat-conduction coefficient of checkerbricks, as related to reversal period and to thickness of bricks Note that k_w is not the overall coefficient of heat transfer

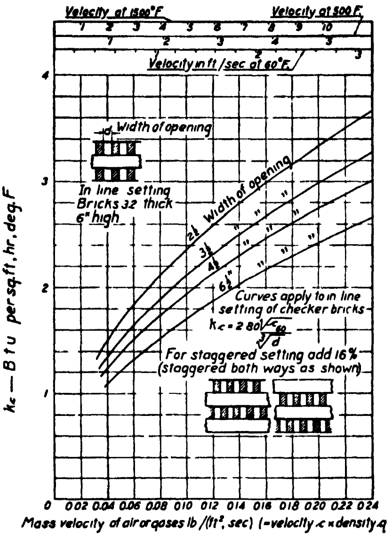

Fig. 165. Test data on heat transfer by convection, in regenerators (radiation not included)

The convection coefficient for checkerwork differs, naturally, from that for recuperator tubes, except in "solid-chimney" checkers. From tests of the experimental regenerator of the German Steelworks Society, Kistner* found that the convective heat transfer is very nearly proportional to the square root of the velocity, and inversely to the cube root of the width of openings between the bricks. The experimental values are shown by the curves of Fig. 165 for checkers with the bricks set vertically in line. For double-staggered setting, the coefficients are increased 16 per cent. All of Kistner's tests were made with bricks 3.2 in. thick and 6 in. high, but they should also be applicable without appreciable change to other usual proportions of bricks. The same values were found to apply to both air and waste gases, and they were practically independent of temperature (for a given mass velocity).

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* *Mitteilungen 139 der Wärmestelle des Vereins deutscher Eisenhüttenleute.*

For "solid-chimney" checkers, the curves of Fig. 150 should be used.

In addition to heat transfer by convection, the radiation heat transfer must be taken into account. In the case of air, there is some absorption of radiated heat by the small amounts of water vapor and carbon dioxide present in the atmosphere, but as shown by the lowest curve of Fig. 43 this is very small indeed. Hence the heat transmission from brick to air depends almost entirely on convection. The heat transfer from the waste gases to the bricks, on the contrary, depends chiefly on radiation from the gases, while convection (for usual widths of openings in the checkers) plays a minor part. The curves of Fig. 43 apply to regenerators as well as to recuperators, but according to Kistner's tests the absorption coefficient by which they must be multiplied is only 70 per cent, for checkerbricks in the standard setting as in Fig. 173.

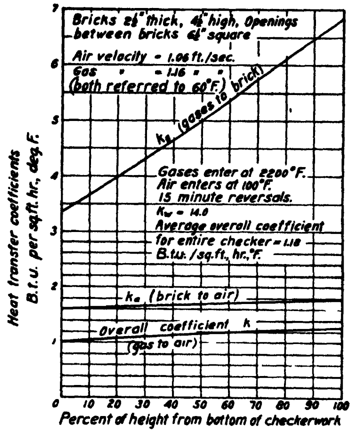

Fig. 166. Variation of heat-transfer coefficient along the height of a typical regenerator.

Figure 166 shows how the coefficients vary, from top to bottom of a typical regenerator. The following example will show the method of calculation:

Example. In the regenerators of a forge furnace (Fig. 157) the checkerbricks are 2½ by 4½ by 9 in., set in line with openings 6½ in. square. Checker chamber dimensions, inside, 10 ft 1½ in. wide by 18 ft 0 in. long. Air quantity = 100 cfs, quantity of gases = 110 cfs, both referred to 60 F. Time between reversals = 15 min. Gases enter the top of the checker at 2200 F. Find the heat-transfer coefficient at that place.

$$\text{Mass velocity of air} = \frac{100 \times 0.0764}{10.12 \times 18 \times \left(\frac{6\frac{1}{2}\,\text{in.}}{9\,\text{in.}}\right)^2} = 0.081 \text{ lb per (sq ft, sec)}$$

From Fig. 165, for 6½-in. openings the convection coefficient is 1.53, and, from Fig. 43, the radiation coefficient for air at a probable temperature of 1800 F is 0.25. Hence $k_a = 1.53 + 0.25 = 1.78$ Btu per (sq ft, hr, F).

Mass velocity of flue gases (of practically the same specific gravity as air) = 110/100 × 0.081 = 0.089 lb per (sq ft, sec). From Fig. 165, the convection coefficient for gases = 1.60; and, from Fig. 43, the radiation coefficient for flue gases at 2200 F in 6½-in. openings is 7.5 Multiplying the latter by the absorption coefficient of 70 per cent gives 5.25, hence,

$$k_g = 5.25 + 1.60 = 6.85 \text{ Btu per (sq ft, hr, F)}$$

From Fig. 164, the brick coefficient for 2½-in. brick thickness, 15-min reversals is 14. Then the overall coefficient of heat transmission from gas to air, at the top of checker, is

$$k = \frac{1}{\frac{1}{6.85} + \frac{1}{14.0} + \frac{1}{1.78}} = 1.28 \text{ Btu per (sq ft, hr, F)}$$

In an actual regenerator there are numerous factors that may and often do modify the coefficients as calculated or as determined from tests of experimental regenerators. One of these factors, as mentioned under "Recuperators," is after-burning. Combustion is frequently incomplete when the gases enter the downtakes to the checkers, and the mixing due to change of direction, together with infiltration of additional air which occurs if the downtakes and checker chambers are not absolutely tight, enables combustion to proceed in the checkerwork, resulting in an apparent rise of the coefficient of heat transfer owing to the increase of temperature in the middle of the checker height. If the flame is luminous in the checkers, an actual increase of the coefficient occurs, even if no after-burning takes place. However, the effect is small.

An influence that very frequently is present, and that largely explains the inefficient heat transfer in many regenerators, is uneven distribution of air and gases, or "channeling," which is explained in Chapter 8 (page 394). In checkers in which the fluid being cooled flows downward while the fluid being heated flows upward, the self-regulating effect (page 394) tends to equalize the flow, and the channeling due to inertia of the air and gases is much less serious than would be expected from tests of flow distribution in checkerwork tested cold; nevertheless, it does exist, and to a sufficient extent to explain the low coefficients often found in regenerators.

At this point it may be mentioned that the flow in horizontal checkers is usually unstable and not self-regulating. For that reason, furnaces are seldom equipped with horizontal checkers. Auxiliary or supplementary horizontal checkers are sometimes built into the flues between main-checkers and reversing valves.

Accumulation of dust on the bricks and slagging of the brick surfaces cause the efficiency of heat transfer to decrease as the furnace becomes old. No definite information is available on the conductivity of dust deposits; in any case, the effect of such deposits must depend on their thickness. The authors estimate that the overall coefficient may be reduced as much as 20 per cent by the insulating action of dust deposits. Slagging, which occurs only on the upper courses of checkerbricks, reduces the capacity of the surface for absorbing radiation. The few data available seem to show that gas radiation may be reduced 10 per cent thereby, resulting in a reduction of 9 per cent in the overall coefficient.

Heat losses from the walls of the regenerators, and leakage of air into them during the gas period and out of them during the air period, also cause an apparent decrease of the heat-transfer coefficient, if the latter is based on the usual method of calculation which neglects these two influences. The method of calculation taking account of the wall losses, as derived by the senior author, is given in the Appendix (page 468). For estimating the effect of leakage, the method derived by Heiligenstaedt for recuperators (Appendix, page 467) may be used; it would be strictly correct, however, only if the inward leakage during one period were equal to the outward leakage during the other period, and if both were evenly distributed over the height. These conditions are not likely to be met. In any case, the question is not of great importance in modern checkers, in which every effort is made to keep the checker chambers, downtakes, and ducts tight at all times.

To some extent, the surfaces of rider walls and the interior surfaces of ducts and downtakes act as additional heating surface, whereby the heat-transfer coefficient, if based on the surface of the checkerwork only, is apparently increased.

The net effect of all these conflicting influences is to cause the overall heat-transfer coefficient, for ordinary checkers operating with natural draft, to range between 0.9 and 1.5 Btu per (sq ft, hr, F). In the authors' experience, higher values (up to 2.5) have sometimes been encountered, but this apparently was due to after-burning. The first figure is usually the lower limit except when channeling and leakage are very bad. (These limits do not apply to metallic regenerators as described on page 251.)

Since the calculation of the heating surface in a regenerator can be reduced to that of a recuperator, the curve sheets (Figs. 154 to 156) can be used by multiplying the heating surface obtained from them by the ratio

$$\frac{\text{Recuperator transmission coefficient}}{\text{Regenerator transmission coefficient}}$$

EXAMPLE. To find the regenerator heating surface required to preheat air to 1600 F for a furnace requiring 1,600,000 Btu per hr useful heat plus radiation and convection losses. Fuel, natural gas; furnace temperature, 2200 F. The surface that would be required if the heat-transfer coefficient were 2.5 would be, from Fig. 156, 203 × 105% = 213 sq ft per 1,000,000 Btu per hr. If the regenerator has the usual flue and brick proportions, the average coefficient, from Fig. 166, is 1.18 Btu per (sq ft, hr, F). The required regenerator surface would then be

$$\frac{2.5}{1.18} \times \frac{1,600,000}{1,000,000} \times 213 = 723 \text{ sq ft}$$

This is the heating surface in each regenerator (not in both).

After the determination of the required area of heating surface, the next step is to decide on the best arrangement and proportions of checkerwork to give this area. The chief variables are brick thickness, and size of flues or passages between the bricks. As far as concerns heat transfer and storage, the thinner the bricks, the better is their efficiency. The center of a thick brick is very imperfectly utilized, first, because its temperature fluctuation is small compared to that of the surface, and, second, because there is a phase lag. This is clearly shown by the curves of Figs. 167 and 168, which were computed by means of Fourier's equations on the basis of a constant gas temperature during one-half of the cycle; a constant (400 F lower) air temperature during the other half; and a mean coefficient of heat transfer for the cycle. In these illustrations the notations $t = 4$, $t = 6$, etc., mean that 4 min, 6 min, etc., have elapsed from the time of reversal. The phase lag is very evident in the 4½-in. brick, where, from $t = 0$ to $t = 6$, the temperature of the center is falling while that of the surface is rising.

This condition leads to the conception of "brick saturation," which indicates the ratio of heat absorbed in a brick to that which would have been stored up if the whole brick had attained the temperature of the surface. For equal brick saturation, the time between reversals should increase approximately as the square of the brick thickness.

From the temperature-space diagrams obtained by the graphical method as mentioned on page 444, and similar to Figs. 167 and 168

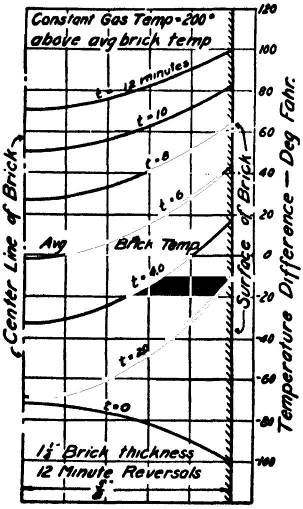

Fig. 167. Calculated temperature distribution in a regenerator brick of 1¼-in. thickness, with 12-min reversals.

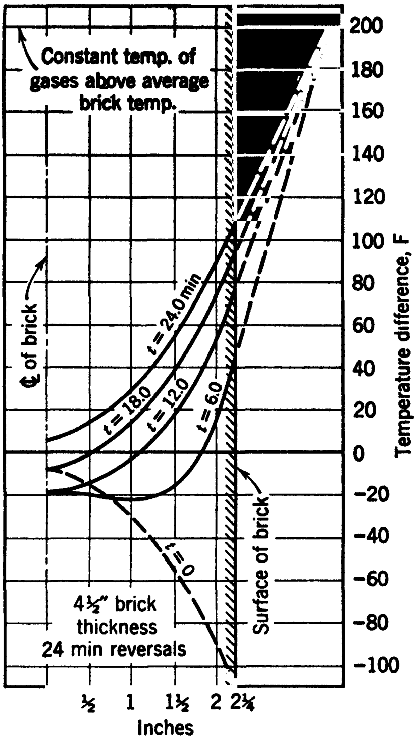

Fig. 168. Calculated temperature distribution in a regenerator brick of 4½-in. thickness, with 24-min reversals.

except that the much higher coefficient of heat transfer from the flue gases to the bricks has been taken into account, the "brick saturation" or heat-storage factors plotted in Fig. 169 were obtained. These calculated factors check almost exactly with the values obtained by Schumacher in tests of the experimental regenerator, when the experimental values are corrected for the different brick thickness. The values of Fig. 169 can therefore be used with confidence.

Although it would be possible, by lengthening the period between reversals, to obtain as high a heat saturation in thick bricks as in thin ones, this procedure would result in too great a drop of the air temperature from the beginning to the end of the period, with consequent undesirable fluctuation of the furnace temperature, unless an inordinately large weight of bricks were provided in the checkers.

With common dimensions of checkerbricks and common time intervals between reversals, fluctuation of temperature of preheated air is so small that it has no effect on furnace operation. In any event, equipment for temperature control compensates for such fluctuation. For these reasons, the extent of temperature fluctuations is not discussed here. In blast-furnace stoves, the interval between reversals is rather long, and the temperature fluctuation becomes quite an item.

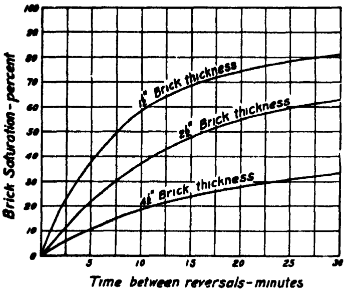

Fig. 169. Saturation of fireclay checkerbricks as related to period of reversal and to thickness of bricks.

Since blast furnaces are outside the scope of this book, the method of calculating the fluctuation of temperature of preheated air is omitted in this edition.

It can be shown by a simple calculation that, for a given brick thickness, the heating surface per cubic foot of checker volume reaches a maximum when the width of flues or passages is approximately equal to the brick thickness; also that, for any given ratio of passage width to brick thickness, the heating surface per cubic foot of checker is inversely proportional to the brick thickness. Furthermore, the smaller the openings the higher is the overall heat-transfer coefficient, for, whereas the transfer from the gases by radiation is reduced when the passages are made narrower, the transfer by convection is increased. As far as the flue gases alone are concerned the two effects nearly balance and the size of the passages makes little difference, but the heat transfer from the bricks to the air, which is effected almost entirely by convection, is increased by reducing the size of the passages. Since in the usual checker the rate of transfer to the air rather than from the gases is the limiting feature, the net result of reducing the passage width is an increase of the overall rate of heat transfer. This, with the increased heating surface, results (other things being equal) in higher temperature of preheat. (This statement as to the heat-transfer rate does not refer to "gas checkers" heating producer gas.) The ideal arrangement, then, strictly from the standpoint of heat transfer, would consist of 1¼-in. bricks with openings not over 1¼-in. wide. But on account of rapid clogging with dust, such narrow openings would be impracticable. Even if the small passages are not clogged, the pressure drop through them may be so great that, with only natural draft available, not enough air can be forced through the checkers to supply the furnace requirements (see Chapter 8). The influence of size of flues as well as of brick thickness on area of heating surface for a given regenerator volume can be judged from a study of Table 18.

TABLE 18. Effect of Size of Regenerator Flue and of Brick Thickness, Bricks 4½ in. Deep

Side of Flue, in.	Thickness of Brick, in.	Cross Section of Unit Cell, sq ft	Heating Surface per Cell and per ft Length, sq ft	Heating Surface per cu ft of Checkerwork (Brick plus Open Spaces), sq ft	Weight of Fireclay Bricks per cu ft of Checkerwork, lb	Ratio of Free Area to Total (Plan) Area
			Standard Checkers (Fig. 173)			
2½	1¼	0.0976	0.682	7.00	42.0	0.444
	2½	0.173	0.949	5.48	62.5	0.250
3½	1¼	0.156	0.873	5.59	33.1	0.546
	2½	0.250	1.163	4.64	52.1	0.340
4½	1¼	0.230	1.063	4.61	27.4	0.613
	2½	0.342	1.378	4.04	44.8	0.412
	4½	0.562	1.870	3.32	62.5	0.250
5½	1¼	0.315	1.255	3.98	23.3	0.664
	2½	0.445	1.593	3.57	39.1	0.472
	4½	0.693	2.066	2.98	56.3	0.302
6½	1¼	0.416	1.445	3.46	20.3	0.705
	2½	0.562	1.805	3.21	34.7	0.520
	4½	0.840	2.380	2.83	51.1	0.349
			Solid Chimney Checkers (Fig. 174)			
4½	1¼	0.230	1.500	6.51	48.4	0.613
	2½	0.342	1.500	4.38	73.4	0.412
5½	1¼	0.315	1.833	5.80	42.0	0.664
	2½	0.445	1.833	4.10	65.8	0.472

Quite evidently, of the arrangements shown, the 2½-in. flue with 1¼-in. tile results in the maximum heating surface per square foot of cross-sectional area, and in the cheapest regenerator, if ordinary rectangular bricks are used. In all cases where the gases are fairly clean, and where 1¼-in. tile is available, these dimensions will doubtless be used in the future, particularly for small and medium-sized regenerators.

One reason why the 1¼-in. tiles are not more frequently used is their lack of stability,* or tendency to fall over sidewise, unless laid as "solid chimney" checkers (page 249). If the edge surfaces were ground flat, their stability would be sufficient; but actually these surfaces are always uneven. For large checkers, as long as we stick to standard rectangular bricks, 2½-in. bricks will be used, both for the sake of strength and because with the long periods of reversal (necessitated for the reasons explained on page 231) a sufficiently high degree of saturation can be obtained with them, while, as shown by the sixth column of Table 18, for a given flue size the volume of checker chamber required to contain a given weight of bricks is less than with the smaller thickness of brick. The use of very wide flues apparently is just a habit of the designer, carried over from open-hearth furnace and possibly glass-tank practice. In these furnaces, enormous quantities of sintered dust are deposited in the checkers by the gases, and wide flues are an absolute necessity; otherwise the checkers would clog up in short order. In heating furnaces, on the contrary, much less dust is carried by the gases, and such wide flues are not necessary. On the other hand, if they are made too narrow, even a small deposit of dust will increase the pressure drop so greatly that not enough air can be supplied by the checkers. The designer is sharply limited in the proportions he can use, so long as only natural draft is available (see Chapter 8).

From the tabulation it may be seen that, within the limits of good design, the heating surface per cubic foot of volume of checkerwork varies between 2.83 and 7.00 sq ft. It follows that the regenerator volume for a given heating surface must vary in the same ratio. For the same reason, the rather common method of comparing regenerator sizes by the "cubic feet of checker chamber per square foot of hearth area of furnace" is not logical, unless brick filling of the same kind and size is always used.

In conjunction with the heating-surface values from Figs. 154 to 156, the values in column 5 of the tabulation—heating surface per cubic foot of checkerwork—really determine the volume of the checker chamber. The only remaining uncertainty is the distribution of that volume over the two factors, cross-sectional area and depth. In Chapter 8, it is shown that uniform distribution of flow of gases over the cross section demands deep regenerators of small cross section. The depth is, as a rule, sharply limited either by excessive cost of excavation and foundation, or by the depth at which the groundwater level is reached. With the depth determined by these considerations,

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* The lack of stability is very annoying during the laying-up of the checkers, even with standard bricks, because the arches on which the checkers rest are seldom level and plane.

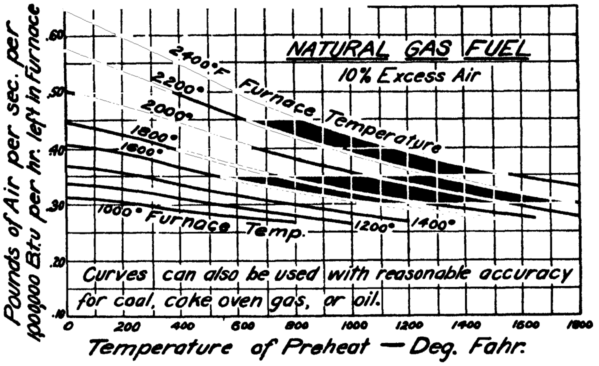

Fig. 170. Air flow through unit-size furnace, using rich fuel.

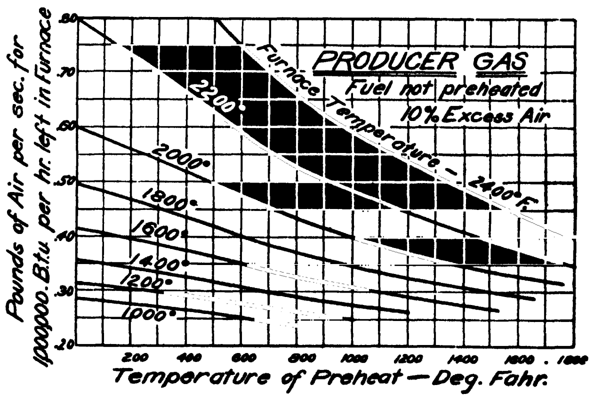

Fig. 171. Air flow through unit-size furnace, using lean fuel.

the cross-sectional area is readily found as previously indicated. The following examples will illustrate the methods.

Since, in the calculation of regenerators, it is necessary to know the quantity of air that is to be heated, whereas usually only the heat consumption of the furnace is known, Figs. 170 and 171 are offered, from which the air quantity corresponding to a given heat consumption can be found easily.

Example 1. A furnace, of which checkers are shown in Fig. 157, requires 17,000,000 Btu per hr useful heat plus radiation and convection losses. The waste gases leave at 2200 F. The furnace was originally designed for natural gas, with provision for the use of producer gas when natural gas should be exhausted. Coke-oven gas is used at present, both checkers being used to preheat air only. Find the temperature of the preheated air.

(*a*) From Table 18, the ratio of free area (in horizontal plane) to total area of checker chamber = 0.520. Actual free area then = (6.75 + 10.12) × 18.0 × 0.520 = 158 sq ft. Trial temperature of air preheat is taken as 1600 F. From Fig. 170, weight of air required = 0.32 lb per sec for each million Btu per hr × 17,000,000/1,000,000 = 5.44 lb per sec. Mass velocity of air in checkers = 5.44/158 sq ft = 0.0345 lb per (sq ft, sec). From Fig. 165, the corresponding convection coefficient from brick to air, for 6½-in. flues, is 1.03. From Fig. 43, the radiation coefficient for air = 0.15. Hence $k_a = 1.03 + 0.15 = 1.18$.

From Fig. 115, 1.082 lb of flue gases are produced per lb of air; the gases have a specific heat about 26 per cent higher than the specific heat of air. Then the final temperature of the gases,

$$T_{2g}, \text{ is } 2200\text{ F} - \frac{(1600\text{ F} - 60\text{ F})}{1.082 \times 1.26} = 1075\text{ F}.$$

From Fig. 43, the gas-radiation coefficient for 6½-in. flues = 7.4 at 2200 F (top of checker) and 2.85 at 1075 F (bottom of checker), with an average of 5.12. But as noted on page 216, for coke-oven gas the radiation is 10 per cent greater; on the other hand, the coefficient must be multiplied by the radiation-absorption coefficient, which Kistner found to be 70 per cent for checker-bricks.

$$110\% \times 70\% \times 5.12 = 3.95 \text{ Btu per (sq ft, hr, F) by radiation}$$

The mass velocity of the gases passing through the checkers is 1.082 × 0.0345 = 0.0374, and, from Fig. 165, the convection coefficient in 6½-in. flues is 1.07. Hence $k_g = 1.07 + 3.95 = 5.02$ Btu per (sq ft, hr, F). From Fig. 164, for 2½-in. bricks and 15-min reversals $k_b = 14.0$. Hence the average overall heat-transfer coefficient for the regenerator is

$$\frac{1}{\frac{1}{5.02} + \frac{1}{14.0} + \frac{1}{1.18}} = 0.896 \text{ Btu per (sq ft, hr, F)}$$

From Table 18, the heating surface per cubic foot of checkerwork is 3.21 sq ft, or (6.75 + 10.12) × 18.0 ft × 9.625 ft depth × 3.21 = 9400 sq ft = heating surface of checkers at each side of furnace.

(*b*) Equivalent heating surface corresponding to the values of Fig. 156 is

$$\frac{1{,}000{,}000}{17{,}000{,}000} \times \frac{0.896}{2.50} \times 9400 \text{ sq ft} = 198 \text{ sq ft}$$

Reading from Fig. 156, the preheat corresponding to this area per 1,000,000 Btu per hr is 1580 F. The trial temperature was close enough to this result that the difference would not cause any appreciable change in the coefficient.

These figures apply if the air and flue gases are perfectly clear and dust-free. Actually, some flame will pass down into the checkers; k will be increased thereby, and preheat will be somewhat higher than that shown. This illustrates a somewhat self-regulating feature of regenerators, namely, that, as the furnace is driven at higher and higher rates, more flame passes down into the checkers and heat transmission becomes more effective as required.

There is another factor, that tends to reduce preheat temperature, namely, the heat loss from the surfaces of the flues leading from the regenerator to the furnace. This action cools the flue gases so that they enter the checkers at a lower temperature than that of the furnace,* and also cools the air going to the furnace. The amount of heat loss can be calculated with the aid of Fig. 86. As mentioned previously, however, the inner walls of the flues or ducts act to some extent as additional regenerator surface.

For calculation of effect of leakage and of heat losses from walls, see Appendix (pages 467 and 468).

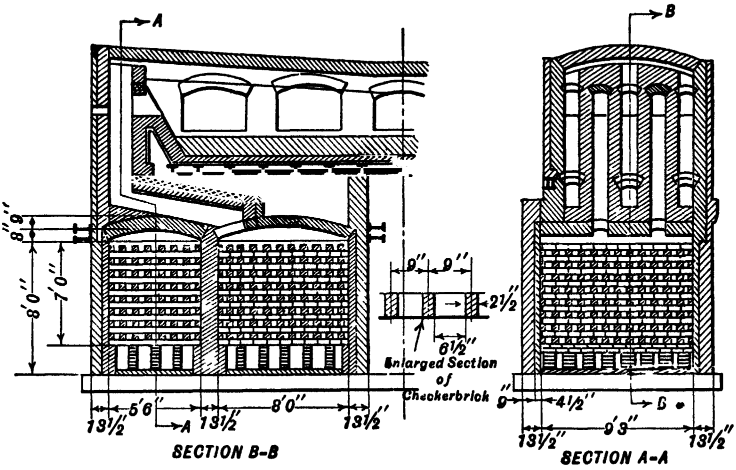

Fig. 172. Regenerative furnace, designed for producer gas.

Example 2. Bridge furnace, shown in Fig. 172, requiring 5,100,000 Btu per hr (useful heat, plus radiation and convection losses). Fuel, producer gas; furnace temperature, 2200 F. Heating surface in air checker, 1670 sq ft, in gas checker, 1150 sq ft, cross-sectional area of passages in air checker, 38 sq ft, in gas checker, 26 sq ft. Width of passages, 6½ in.; length, 7 ft. Find the temperature of preheat of the air.

This example differs from the preceding one in that the fuel gas is preheated.

(*a*) In many of the older furnaces, separate checkers were provided, with the idea of preheating the producer gas. As the gas usually enters from the producers at a fairly high temperature, and as only a small checker is provided, the preheating, as far as increase of temperature is concerned, is not very effective. It has, however, the effect of cracking hydrocarbons and producing a more luminous and radiant flame. If the producers are located at a considerable distance from the furnace, preheating of the fuel gas is effective also with regard to temperature rise. The fuel-gas temperature at the entrance to the furnace is about the same as or slightly higher than the air temperature.

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* In high-temperature melting furnaces, the flues between furnace and regenerators are purposely built thin-walled. The heat loss reduces the temperature to such an extent that outgoing slag particles are solidified before they reach the regenerator.

Figure 171 applies where the fuel is not preheated; it can be used where fuel is preheated by considering the air as heated to a proportionately higher temperature. Trial temperature of preheat = 1200 F; from Fig. 116, the sensible heat in the gas (fuel) at a given temperature is about 89 per cent of that in the combustion air at the same temperature. Then, the equivalent temperature of preheat of the air, if the fuel were not preheated, would be approximately $1200\text{ F} + 0.89 \times (1200\text{ F} - 60\text{ F}) = 2210\text{ F}$. Extrapolating curves of Fig. 171 gives 0.32 lb of air required per sec per 1,000,000 Btu per hr left in the furnace by the gases.

$$\frac{5{,}100{,}000}{1{,}000{,}000} \times 0.32 = 1.64 \text{ lb per sec}$$

Mass velocity of air in checkers = 1.64/38 = 0.043 lb per (sq ft, sec). From Fig. 165, $k_c = 1.13$ Btu per (sq ft, hr) for the air. From Fig. 43, $k_r = 0.15$ for air; hence $k_a = 1.13 + 0.15 = 1.28$.

The waste gases enter at about 2100 F and leave at about 1100 F. The quantity of gases going to the air checker is about 60 per cent of the total. Since 1.76 lb of waste gases are produced per lb of air, mass velocity

$$= \frac{1.64 \times 1.76 \times 60\%}{38} = 0.046 \text{ lb per (sq ft, sec)}$$

in the air checker. Then k_c for gases is 1.16 (Fig. 165). At the top, k_r is read from Fig. 43 as 7.0, and, at the bottom, as 3.0, with an average of 5.0. But the value must be reduced 8 per cent for the products of combustion of producer gas (page 216) and multiplied by the radiation-absorption coefficient of the brick surfaces, 0.70. Hence $k_r = 0.70 \times 0.92 \times 5.0 = 3.22$ for the gases, and $k_g = 3.22 + 1.16 = 4.38$ Btu per (sq ft, hr, F). $k_w = 14.0$ as before; hence the average overall coefficient of heat transfer from the gases to air is

$$\frac{1}{\dfrac{1}{4.38} + \dfrac{1}{14.0} + \dfrac{1}{1.28}} = 0.93 \text{ Btu per (sq ft, hr, F)}$$

Equivalent heating surface per 1,000,000 Btu per hr, for Fig. 155, is

$$\frac{0.93}{2.50} \times \frac{1{,}000{,}000}{5{,}100{,}000} \times 1670 = 122 \text{ sq ft}$$

Reading from Fig. 155, for 2100 F temperature of gases and 122 sq ft area, the temperature to which the air is heated is found to be 1120 F. This is somewhat lower than the assumed temperature of preheat, but not sufficiently different to require changing the coefficients.

In the preceding examples, it has been assumed that the bricks were laid in the "standard" manner, as shown in Fig. 173. In practice, several other arrangements are also in use, as indicated in Figs. 174 and 175. Each method of laying has its adherents, and vigorous arguments are frequently heard about the relative merits of increased heating surfaces, breaking up of flow, entrance of heat from four sides instead of two, etc. As shown by Table 18, the "solid-chimney" checker shown in Fig. 174 has more actual area of heating surface and

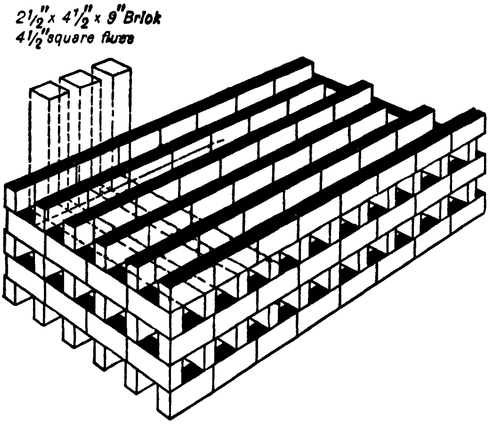

Fig. 173. Standard checkerwork for regenerators.

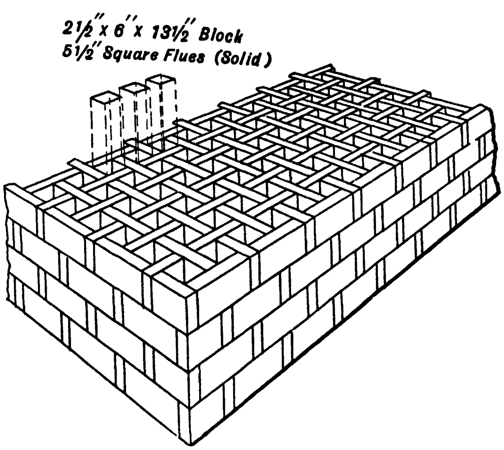

Fig. 174. Solid chimney checkers.

a greater weight of brick per cubic foot of checkerwork than has the standard checker of Fig. 173. The gas-radiation coefficient should be somewhat higher in the solid-chimney type, because radiation from the flowing gases is normal to all the surfaces and is more effective than radiation at an angle to the horizontal surfaces of the standard checkerbricks; on the other hand, the convection coefficient is certainly lower in the solid-chimney type, as may be seen by comparing

Fig. 150 with Fig. 165; the discontinuities in the passages of the standard checker cause and maintain turbulence of flow and thereby increase the heat transfer by convection. Apparently, all these conflicting influences just about balance, for the authors' close observations of different checkers in practice, under otherwise equal conditions, have failed to show any advantage of one method over the others: for a given volume of checkerwork and with the same thickness of bricks the preheat temperature is about the same for all three types of setting.

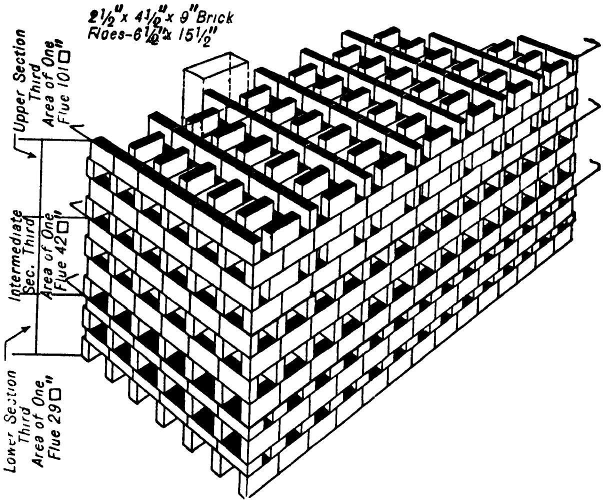
2½" x 4½" x 9" Brick
Flues-6½" x 15½"
Upper Section Third Area of One Flue 101□"
Intermediate Sec. Third Area of One Flue 42□"
Lower Section Third Area of One Flue 29□"

Fig. 175. Checkerwork with variable cross section of flue.

Many different settings of bricks have been devised for the purpose of increasing heat transfer in regenerators. For the same purpose and also for increasing stability, new shapes of bricks for regenerators have been designed. Among the many inventions, only those are of value that increase the area for flow from the bottom of the regenerator to its top, while keeping the heat-exchanging surface constant. In view of the limited use of regenerators in industrial furnaces, the various arrangements are not described here.

Continuous Regeneration. Alternation of flame direction in furnaces is often inconvenient and is sometimes impossible, as, for instance, in blast furnaces. For both reasons, several designs have appeared for making the furnace recuperative (with constant direction of flame) while making the heat-salvaging equipment regenerative. These designs consist either of a plurality of cyclically operated regenerators or else of a regenerator with movable parts. Blast-furnace stoves are a typical example of the first group. For several years, metallic regenerators, working on the same principle, were on the market as heat-salvaging equipment for industrial furnaces. Short life caused their abandonment.

In the second group, the material that alternately absorbs and emits heat is movable. In one design, which is used as an air preheater for steam generators, a ring, having a vertical axis rotates slowly. The ring is filled with thin, corrugated sheets of heat-resisting metal. Hot products of combustion pass down through one half of the ring while air flows upward through the other half. This equipment is too delicate for use with industrial furnaces. Another type of continuous regenerator is the pebble heater. Pebbles or marbles of refractory material flow downward through a funnel to a throat, where they meet hot gases. Below the throat, the pebbles heat upwardly flowing air. The cold pebbles which are discharged at the bottom are carried back to the top by a conveyor. The pebble heat exchanger has given a good account of itself in some processes. So far as the authors know, it has not been applied to industrial furnaces.

(e) Waste-Heat Boilers. The third method of utilizing the heat of the products of combustion, mentioned on page 205, is the generation of steam. A brief discussion of waste-heat boilers, in connection with heating furnaces, is appropriate.

The amount of steam that can be generated in unit time from a given furnace depends on the weight of the flue gases in unit time, their temperature at entrance to the boiler, and the temperature to which they can be cooled. The temperature of the gases at entrance to the boiler should be approximately equal to the furnace temperature, except when the boiler is used in connection with a continuous furnace, in which case the flue-gas temperature depends on the rate of heating, as was explained in Chapters 4 and 5. In heating-furnace practice, waste-heat boilers are practically never used if the furnace is equipped with either a regenerator or recuperator.

The flue-gas flow depends on the heat requirements of the furnace and its temperature; it can be found by the methods explained earlier (see pages 184 to 200 and 245). The temperature of the hot flue gases entering the boiler may slightly exceed the temperature of the furnace, if the boiler is mounted immediately above the furnace and is connected with the latter by a very short flue; it will be lower than the furnace temperature if the boiler is set some distance away from the furnace. For the purpose of computing a waste-heat boiler chart, the assumption will be made that the products of combustion reach the boiler with furnace temperature.* The temperature of the stack gases at the place where they leave the boiler depends on rate of gas flow, temperature of entering gases, heating surface of boiler, steam and water temperature, and coefficient of heat transmission. The latter varies with the temperature of the gases, the cleanliness of the boiler surface (inside and out), and the arrangement of the passes. For the purpose of calculating charts, three stack temperatures were assumed, namely 500, 750, and 1000 F.

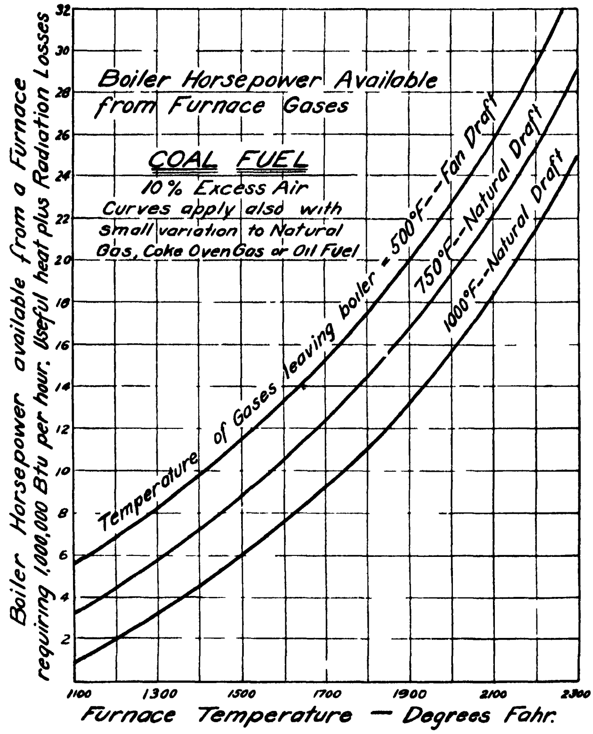

Fig. 176. Boiler horsepower available from unit-size furnace under different conditions. One boiler horsepower equals 33,475 Btu per hr.

Two types of graphs were computed. The first type (see Fig. 176) shows the number of boiler horsepower (at 33,475 Btu per hr) that can be extracted from the products of combustion of a furnace rated at 1,000,000 Btu per hr with varying furnace temperatures and with three stack temperatures. The second type of graph (Fig. 177) gives the heating surface that is necessary per boiler horsepower with different furnace temperatures and three stack temperatures.

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* Although open-hearth furnaces and glass tanks are outside the scope of this book, it may be mentioned that the waste gases coming from furnaces of either of the two types reach waste-heat boilers at less than regenerator-discharge temperature. Although the temperature in open-hearth furnaces equals 3000 F, the temperature of the waste gases at entrance to the waste-heat boiler is only 1100 to 1200 F. For glass tanks, the corresponding values are 2600 and 900 to 1000 F.

The values of Fig. 176 were found as indicated previously. For a given furnace temperature, the amount of heat, per unit of fuel, in the flue gases leaving the furnace, and the amount given up in the furnace can be determined by use of the curves (Figs. 109 to 121). If the furnace requires that 1,000,000 Btu be given up per hr by the gases for heating stock and furnace parts, and for heat losses by radiation and convection, then the total amount of heat left in the flue gases per hour can readily be found from the above figures. In the calculation of Fig. 176, 10 per cent excess air was assumed. The ratio of the heat that can be abstracted from the flue gases down to a given leaving temperature to the heat abstracted in the furnace can likewise be found from Figs. 109 to 121; and this ratio multiplied by 1,000,000/33,475 equals the available boiler horsepower. Curves for other fuels of high calorific value, such as oil, natural gas, tar, and coke-oven gas, were also computed, but they are so much like those of Fig. 176 that it is not worth while to reproduce them.

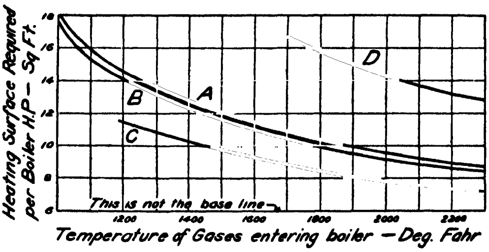


Fig. 177. Heating surface required for waste-heat boilers, with clean surfaces. Curve *A* is for draft loss through boiler of 2-in. water; coefficient of heat transfer, $K = 6.0$; curves *B*, *C*, and *D* are for draft loss $= 0.5$; $K = 4.0$.

A and *D* 500 F temperature of gases leaving boiler.
B 750 F temperature of gases leaving boiler.
C 1000 F temperature of gases leaving boiler.

It is quite apparent that much waste heat is available from 2200 F furnaces, and a comparatively small amount is available from 1400 to 1600 F furnaces, unless fan draft is used and the gases are cooled down to a low temperature.

The temperature of the stack gases, as before stated, is affected by several variables. The relation between the principal variables is shown in Fig. 177. For the calculations, it was assumed that, if a fan is used, a heat-transfer coefficient of 6 is obtained, with a draft loss of 2 in. of water through the boiler. (Calculation of the heat-transfer coefficient by means of the equations of Appendix 4 showed that, for a given draft loss, there is little change of the coefficient with change of temperature.) It was assumed that, with a stack temperature of 750 and 1000 F, natural draft would be satisfactory, and a heat-transmission coefficient of 4 Btu per (sq ft, F) is obtained, with 0.50-in. draft loss through the boiler. Curves *A* (for fan draft and 500 F stack) and *B* (for natural draft and 750 F stock) lie very close together, as might be expected. It is a question of finance whether it pays to install a larger boiler with a natural draft or a somewhat smaller boiler and fan. Further discussion of this question belongs in a treatise on steam boilers and will not be followed here. Curve *C* is for natural-draft gases leaving the boiler at 750 F, $k = 4$; curve *D* is for natural draft, leaving temperature 500 F, $k = 4$.

The use of Fig. 177 is simple. First, find the available boiler horsepower from Fig. 176 and multiply by the number of square feet needed per boiler horsepower from Fig. 177.

Nevertheless, judgment is necessary in the use of Figs. 176 and 177, because they are based on the assumption that *all* the products of combustion reach the boiler *at furnace temperature*. In practice, neither of the two emphasized conditions is realized, with the result that the actual available horsepower is often only a fraction of that indicated by the graphs.

In countries with high fuel cost and low labor cost, even the heat in the water that flows through skid pipes is utilized in waste-heat boilers. In order to prevent deposits of scale in the skid pipes, the circulating water must either be treated or else be condensate. The water is under pressure and may be heated to a high temperature; heating, of course, depends upon the steam pressure in the boiler. With modern boiler pressures, the water may be heated to approximately 500 F. The high water temperature reduces heat loss in the furnace. Steam bubbles accidentally formed in the skid pipes are very small at high pressures and are harmless. The here described method of heat salvage requires a close coordination between furnace operation and power-plant operation.

The economics of waste-heat boilers are discussed in Volume II.

Installations in which a single furnace is equipped with a waste-heat boiler are very scarce. Contrariwise, it is common practice in small forge plants to attach a waste-heat boiler to two or more furnaces. In such plants, the short circuit (required by law in some European countries) from furnace to stack is usually omitted. This short circuit is opened up in case of accident to the boiler and allows continued operation of the furnace. It can also be opened up if there is danger of overheating part of the boiler, and possibly causing an explosion.

In those American plants that use waste-heat boilers, this by-pass is of little use, for, if a boiler tube bursts, the steam hammers or press must stop operating, and it is useless to try to keep the furnace hot. Furthermore, when water-tube boilers are used the danger of explosion is almost entirely absent, and, if a boiler tube bags or bursts and hot gases pass through the boiler for a while, not much harm is done.

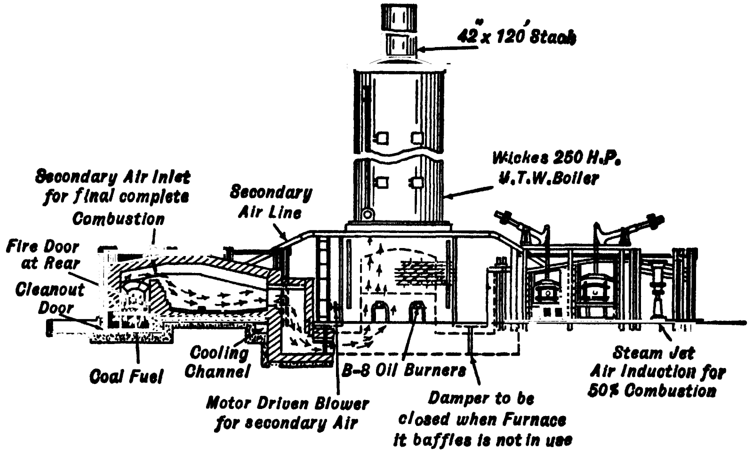


Fig. 178. American installation of waste-heat boiler on forge furnace.

An American installation of this type is shown in Fig. 178. It shows two coalfired furnaces discharging their gases through a Wickes boiler. The flues are very short. Auxiliary oil burners are provided for emergency purposes. Dampers are installed between each furnace and the boiler, and are used either in the wide-open or the dead-shut position. They are necessary because one furnace may be off for repairs. It is also advisable to have a damper on the low-temperature side of the boiler for regulating the pressure in the furnace.

An English waste-heat boiler installation is shown in Fig. 179, and a German installation in Fig. 180. In the latter illustration, the by-pass, with its removable brick partition, is clearly shown. It is obvious that a boiler with flat superheater surfaces, as shown in this illustration, needs quick action in case of overheating.

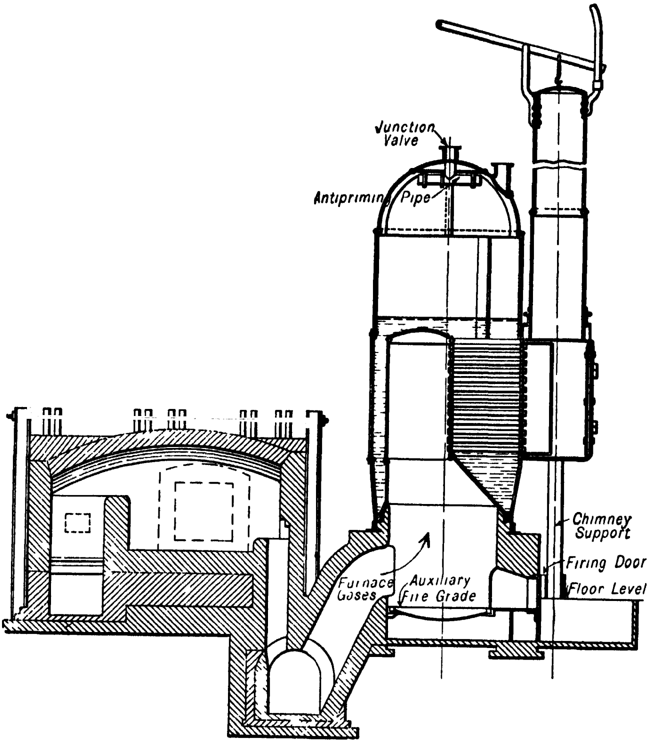
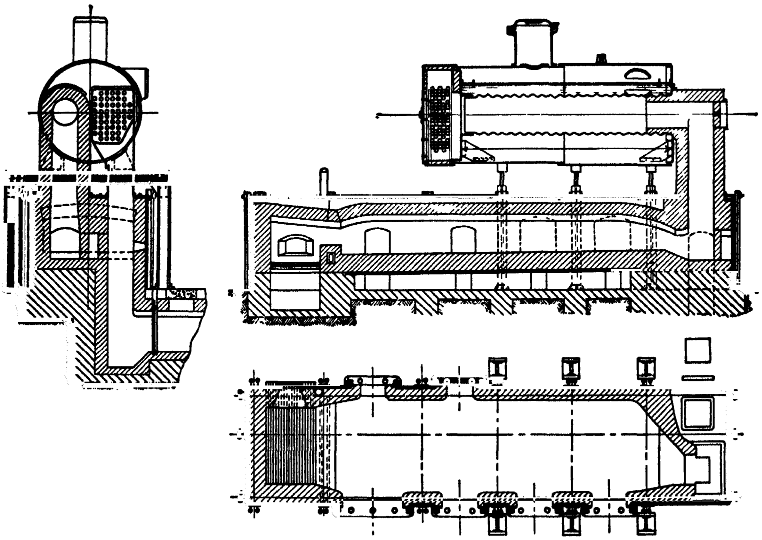


Fig. 179. English installation of waste-heat boiler on heating furnace.



From the foregoing examples it may be concluded that 10 to 90 per cent of the products of combustion may reach the waste-heat boiler, and that a careful weighing of all the circumstances is necessary for a correct judgment of the proper fraction of the products of combustion.

The curves of Fig. 176 may be used where there is an appreciable temperature drop of the gases passing from the furnace to the boiler, by multiplying the values by the factor,

$$0.97\left(\frac{\text{Temperature entering boiler minus temperature leaving boiler}}{\text{Temperature leaving furnace minus temperature leaving boiler}}\right)$$

For instance, if the furnace temperature = 2300 F, temperature drop in flue = 200 F, and gases leave boiler at 750 F, then the coefficient is $0.97\left(\frac{2100-750}{2300-750}\right)=0.84$. Boiler horsepower available per million Btu used by furnace = 0.84 × 29 (from Fig. 176) = 24.4 boiler horsepower.

This is not the place for a treatise on boiler design, but a few words are needed concerning the differences between the usual boiler installation and a waste-heat installation. In the latter, all the heat transfer is effected by convection and by radiation from clear gases, whereas in the former, the greater part is effected by radiation from the flame or fuel bed. In waste-heat boilers, therefore, not only is the heat-transfer coefficient lower, but also the average temperature difference is considerably less, and a larger amount of heating surface is required for a given boiler horsepower output. The heat transfer can be increased and the required heating surface reduced by passing the gases at high velocity through long, narrow paths. This is possible, however, only by the use of fans to supply additional draft; accordingly, the majority of modern installations are designed for induced draft.

For the extraction of waste heat, the single-pass horizontal fire-tube boiler having a very large number of small tubes is now widely used in the United States. An example is shown in Fig. 181. With a given draft, a higher heat-transfer rate can be obtained in this type of boiler than in the water-tube type. Furthermore, there is less danger of explosions where the gases contain unconsumed combustible, and less chance of air infiltration. It is also stated that the fire tubes are kept clean more easily; this is very important in waste-heat boilers, because any reduction of the already low rate of heat transfer by an insulating layer of dust greatly reduces the heat recovery. The fire-tube boiler is not suitable for installations where the temperatures vary rapidly, because tube leaks develop; neither is it desirable where the feedwater is very bad, because of the difficulty of removing scale from it. In such cases, water-tube boilers are used. The two types have been found to have about the same efficiency of heat recovery when the gases are above 1800 F, but at lower temperatures the water-tube type falls behind very rapidly, probably because of air infiltration. A vertical fire-tube boiler is shown in Fig. 182. This waste-heat boiler was developed to serve in connection with water-gas generators.

Fig. 181. Horizontal fire-tube waste-heat boiler.

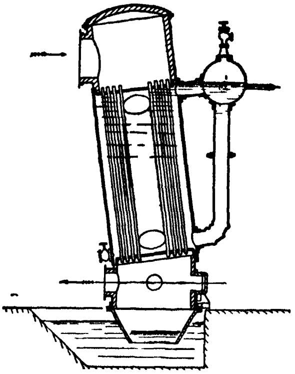


Fig. 182. Vertical fire-tube waste-heat boiler.

In modern installations, an economizer is occasionally installed in series with the boiler, and sometimes a superheater also.

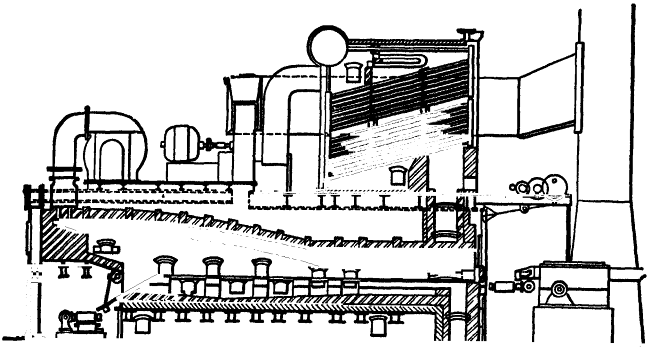


Fig. 183. Waste-heat boiler over continuous furnace.

Mention was made of the possibility of installing waste-heat boilers over hard-driven continuous furnaces. An example of this type is shown in Fig. 183. This furnace and boiler installation was described by O. D. Pratt in a paper before the American Society of Mechanical Engineers, in December 1916. In order to secure high output for the limited floor space, the furnace was purposely designed with a large combustion chamber, and was intended to be "flame-filled." The flue gases leave the furnace with temperatures up to 1745 F, which is very attractive for the generation of steam.

Whenever a continuous furnace is called on to heat thick steel billets or slabs at a high rate (per sq ft, hr), the installation of an overhead waste-heat boiler, as shown in Fig. 183, is correct engineering, provided there is appropriate use for steam reasonably near the furnace.

In the first quarter of the twentieth century, a waste-heat boiler was a good investment, because recuperators were unreliable and because much of the machinery near a furnace was operated by steampower. At this writing, waste-heat boilers are less desirable, because properly designed recuperators are reliable, and, furthermore, machinery is operated by electric motors which receive their energy from a central powerplant. The latter may be several miles away from the furnace.

chapter 7

Strength and Durability of Furnaces

1. PROPERTIES OF MATERIALS USED IN FURNACE CONSTRUCTION

Furnaces are built up of metal parts, of heat-resisting non-metallic materials, which are called "refractories," and of concrete. The factors that affect the strength and durability of furnaces are:

1. High temperature.
2. Variations of temperature.
3. Chemical effect of oxides, sulfur compounds, and slags.
4. Oxygen and other gases or vapors at high temperature.
5. Explosions.
6. Mechanical injury by moving solids.
7. Steam and water.
8. Excessive loading on roofs and sidewalls.
9. Yielding of foundations.
10. Vibrations.

These individual causes produce a variety of effects in hearths, sidewalls, roofs, binding, doors, resistors, recuperators, and regenerators. Before these different effects can be evaluated, the physical and chemical properties of the materials entering into furnace construction must be briefly reviewed.

Evidently, the furnace engineer selects those materials which have the required strength and durability at lowest cost.

Metals are used in or on furnaces for electrical resistors, skids and hearth bars, walking beams, rollers, roller rails, furnace chains, muffles, jackets, binding, furnace fronts, door frames, and sills. Cast iron and carbon steel, being the cheapest metallic materials having great strength, are used wherever possible. They are not durable when exposed to high temperatures, first, because their mechanical strength decreases, and second, because they oxidize and burn away. Figure 184 shows the quick pull tensile strength of soft, medium, and hard steel as a function of temperature; Fig. 185 shows the tensile strength of gray cast iron and of ductile iron. This illustration teaches that at elevated temperatures conventional cast iron should not be used in tension, but only in compression. Ductile gray (nodular) cast iron may be used in tension.*

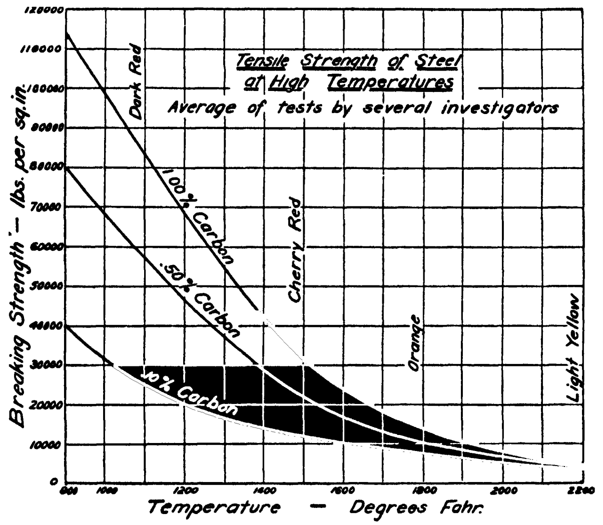


Fig. 184. Tensile strength of steel at various temperatures.

Inside the furnace, that is to say, for hearth bars of annealing furnaces and of heat-treating furnaces, cast iron is better than steel, because it does not oxidize quite so readily. If cast iron is used for hearth bars or hearth plates, it is advisable to subdivide the hearth into short sections and to lock one end of each section into the brickwork, allowing for expansion and growth (see Fig. 186). It may be argued that iron and firebrick have almost the same linear expansion, and consequently no subdivision is needed; but the hearth bars assume

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* Complete information on the properties of cast iron can be found in *Gray Iron Casting Handbook,* published 1958 by the Gray Iron Founders' Society, Cleveland, Ohio.

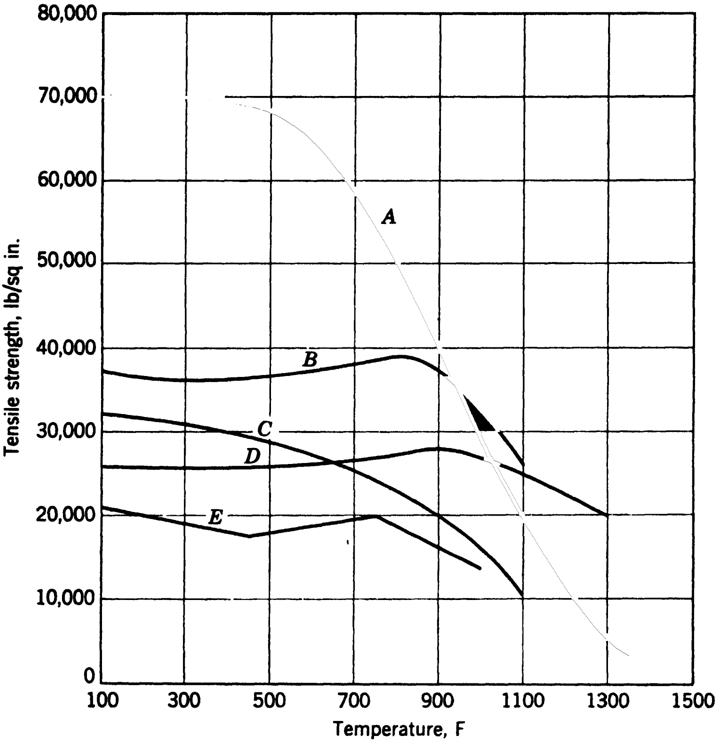

Fig. 185. Tensile strength of cast iron at elevated temperatures.

A—Ductile (nodular) iron, annealed
B—Gray iron as cast
C—Gray iron, annealed 200 hr at 1020 F
D—Gray cast iron
E—Gray cast iron, 3.45% C, 2.1% Si

furnace temperatures much more quickly than the brickwork, and ordinary cast iron grows in volume after each alternate heating and cooling. For both reasons, space must be allowed for expansion. In a few furnaces the chemical composition of cast iron or steel bars is an objection to their use, and other materials must be employed.

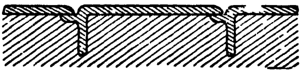

Fig. 186. Cast-iron hearth plates anchored in brickwork.

The growth or irreversible expansion of cast iron becomes noticeable above 800 F. It is considerable after repeated heating to 1500 F or above. There appears to be no limit to the growth of cast iron that is repeatedly heated to 1500 F (or above). Silicon or chromium, or both, greatly reduce the permanent growth of cast iron. Ductile iron grows less than ordinary iron. Figures 187 and 188 contain reliable information on the growth of cast iron.

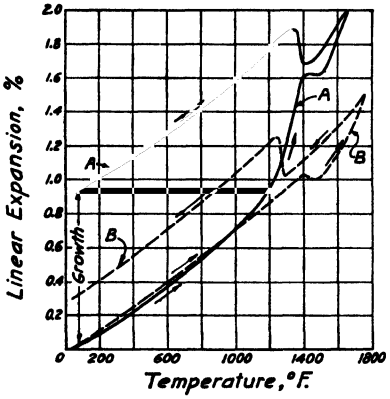

Fig. 187. Expansion and growth of cast iron, after a single heating.

Curve *A* for cast iron of 3.08 per cent C, 1.68 per cent Si. (Symposium ASTM 1931, p. 689.)

Curve *B* for cast iron of 3.99 per cent C, 1.60 per cent Si, heated in ½ hr, then cooled in 2½ hr. (*Engineering,* 1931, p. 839.)

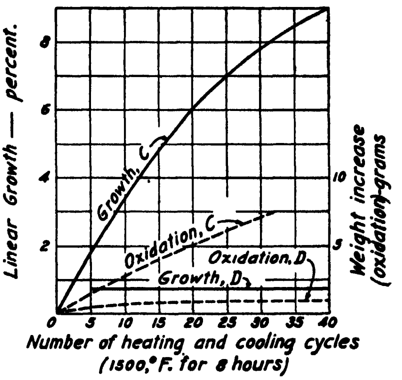

Fig. 188. Growth and oxidation of cast iron after repeated heating and cooling.

Curves *C* for plain cast iron, 3.26 per cent C, 2.02 per cent Si.

Curves *D* for cast iron containing 3.04 per cent C, 1.62 per cent Si, 14.31 per cent Ni, 5.37 per cent Cu, 3.26 per cent Cr. (*Transactions American Society for Steel Treating,* Volume 18, 1930, p. 923.)

Reported tests show a growth of 9 per cent in 40 cycles totaling 320 hours. Other tests have shown a growth for plain cast iron of 3½ per cent for 35 cycles (total 320 hours) at 1472 F. It is evident that repetitive heating as well as temperature are important in the amount of growth, and that ample room must be provided for this growth.

Any cast iron may be used for temperatures below 1300 F. Ductile (alloy) iron is serviceable up to 1600 F. Above this temperature, no cast iron is durable.

Steel also exhibits the phenomenon of permanent growth after repeated heating to 1500 F or above, but the growth is very small in comparison with the permanent growth of cast iron. Growth of steel occurs if great temperature differences exist between surface and interior.

Neither cast iron nor plain carbon steel can be used on the inside of high-temperature furnaces (above 1600 F) unless the metal is water-cooled or unless protective atmosphere gas is used. Dead-soft, welded steel plate is a serviceable material for water-cooled metal parts inside of the furnace, because it does not exhibit cracks if subjected to great differences of temperature. In furnaces with hydrogen and other protective atmospheres, alloy-steel is used for covers and for other metal parts at temperatures up to 2100 F.

Metals that are to have extended life in furnaces with temperatures in excess of 1400 F must meet two requirements:

1. They must not be oxidized rapidly; neither must they be attacked by mildly sulfurous atmospheres.
2. With a reasonable stress, deformation must be so slow that reshaping or replacement becomes necessary only after long usage.

A number of alloying elements, if added in sufficient proportion, impart to iron one or the other or both of these properties. Such elements are chromium, nickel, cobalt, molybdenum, tungsten, columbium, and titanium.

During and after World War II, a large number of heat-resisting alloys were developed. They contain various percentages of the above-named elements and still others. These alloys, which were given numbers, were intended for parts of gas turbines and oil turbines. For that use, resistance to "creep" under high stress and at high temperature is very important because of the high stress which is caused by centrifugal force (which stress cannot be reduced by increase of cross-section) and because of the small clearances between stationary and revolving parts.

Inquiries among furnace builders and makers of heat-resisting alloys indicate that, so far as industrial furnaces are concerned, iron-carbon-chromium-nickel alloys, usually without other alloying elements, are, at this writing, generally used. Among the few exceptions is an alloy (NA-22-H), patented in the United States, that has great strength at temperatures up to 2200 F, and resists oxidation at these temperatures. This alloy, which contains tungsten, is used for furnace rails, radiant tubes, burner nozzles, and other difficult furnace parts. The probable reasons for getting along without the rarer elements are that tensile stress can be kept low and that a noticeable amount of creep does but little harm in industrial furnaces. For the walls of metallic recuperators, a small addition of titanium or columbium is helpful (see section on strength and durability of recuperators, Volume II).

The nickel-chromium alloys are extensively used for electrical resistors, carburizing boxes, heat-treating containers, hearth plates, chains, walking beams, recuperators, and for any other purpose for which resistance to oxidation, coupled with high heat conductivity (compared to refractories), or high tensile strength is needed. The use of nickel-chromium alloys would be even more general if it were not for their price, which is 15 to 20 times that of steel, varying with market fluctuations.

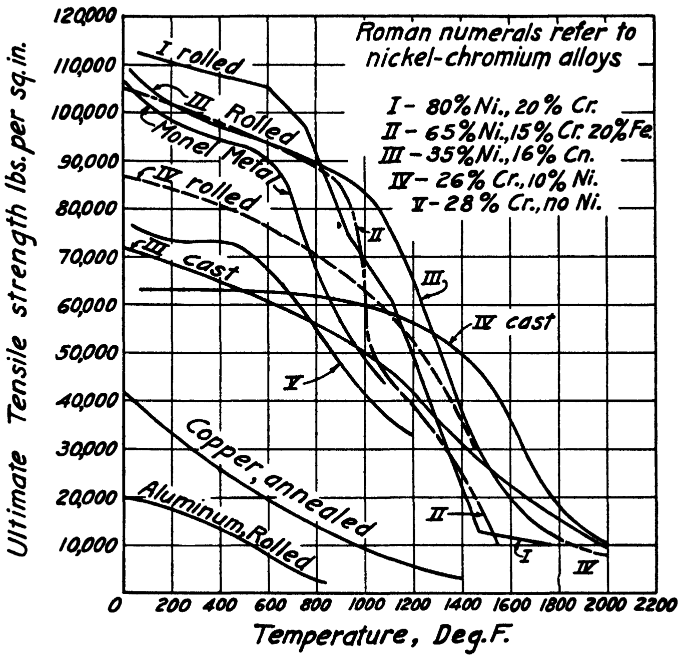

Fig. 189. Temperature-strength relations for various metals and alloys. Quick-pull tests.

There have been few changes in the data on the properties of nickel-chromium alloys since the publication of the fourth edition (1951). Table 19 shows those alloys which are commonly used in furnace design.

In judging the strength of thermoalloys, a distinction must be made between quick-pull strength and creep strength. Hot metals are viscous. This statement means that, although they oppose great resistance to quick deformation, they are deformed slowly and steadily by a stress which is only a fraction of the quick-pull strength. Although the quick-pull strength is of value only for purposes of comparison, data on this strength are given in Fig. 189.

Information on creep strength is more important. By creep strength

TABLE 19. Physical and Chemical Properties of Heat-Resisting Alloys

Group		I	II	III
Designation in U.S.A.		Inconel	"65–15"	NA-22-H
Approximate chemical composition, per cent	Cr	14	11–20	26.3
	Ni	80	60–64	46
	C	0.20 cast 0.08 rolled	. . . . . .	0.44 . . .
	Si	2.0 cast	. . .	0.99
	Mn	0.25 rolled	. . .	1.36
	Fe	6.0–6.5	20–25	. . .
	W	. . .	. . .	5.28
Specific gravity		8.4 . . .	8.29 rolled 8.06 cast	. . . . . .
Weight, pci		0.307	0.290	. . .
Tensile strength, cold, psi		112,000 rolled 70,000–95,000 cast	64,000 cast 102,000 rolled	65,000 cast . . .
Heat expansion per (F, in.) per in.		0.000009	0.0000073	0.0000086
Shrinkage allowance for castings, in. per ft		. . .	. . .	. . .
Heat conductivity, Btu per (ft, hr, F) at room temperature		8.65	6.2	. . .
Specific heat		0.109	0.114 at room temp.	. . .
Electrical resistance, ohms per circular mil ft		620	660 rolled 720 cast	. . .
Temperature coefficient of electrical resistance		0.000006	0.0000095	. . .

is meant that stress which, in a given time and at a given temperature, will not stretch the material more than a given fraction of its original length. It may be seen readily that the creep strength varies not only with the temperature but also with the amount of specific (or unit) elongation in a predetermined time. Commonly accepted time figures for creep strength are 1 per cent stretch in 10,000 hr, and also 1 per cent in 100,000 hr. Figure 190 teaches that, at high temperatures, an exceedingly small stress suffices to bring about a creep of 1 per cent in 100,000 hr. If a furnace is operated 20 hr per day, 300 days per yr, the working hours in 1 yr are 6000. At a deformation rate of 1 per cent in 100,000 hours, the stressed object would be deformed 1 per cent in a time which is longer than 16 yr. Competition among furnace builders has caused them to work with stresses that are much higher

TABLE 19 *(continued)*

IV	V	VI	VII
"35–15"	"25–12"	Type 446	Type 309
15–20	22–30	23–27	22–24
35–40	8–18	0	12–15
0.40–0.60 cast . . .	0.50 cast 0.25 max rolled	0.50 cast 0.10–0.25 rolled	0.05–0.15 max rolled
. . .	1.0–1.5	1.0 max	1.0 max
. . .	1.0	1.5	2.0 max
Balance	Balance	. . .	. . .
. . .	. . .	. . .	. . .
7.92	7.0	7.65	7.88
. . .	. . .	. . .	. . .
0.290	0.252	0.270	0.285
65,000–68,000 cast 95,000 rolled	65,000–110,000 cast 90,000–115,000 rolled	40,000–50,000 cast 75,000–95,000 rolled	95,000 cast 85,000–110,000 rolled
0.0000075	0.000009	0.0000069	0.0000108
¼ in.	¼ in.	. . .	. . .
5.5–7.5	9.4	12.1	8.0
0.112 at room temp.	0.14 mean, 60–1600 F	0.12 mean, 60–1600 F	0.118 at room temp. 0.155 mean (to 1600 F)
670 cast	570	413	481

than those which cause this almost imperceptible deformation. Carbon content in any ferrous alloy increases creep strength but reduces ductility, with the result that the high-strength alloys cannot be forged or rolled. In consequence, rolled parts, such as plates, bars, and tubes have a lower creep strength than high-carbon castings of alloy steel. Figure 190*a* contains information on the creep strength of several commonly used low-carbon alloys. Examples of working stresses for cast alloys are given in the curves (Figs. 191 to 196) which were obtained from the Electro-alloys Division of the American Brake Shoe Company. (Other makers of alloys recommend similar values.) It will be seen that the limiting creep stress is defined as that stress which will produce 1 per cent elongation in 10,000 hours and that the recommended design stress is one-half of the limiting creep stress. Many furnace designers use these values for rotating parts such as rollers, where the stress is continuously reversed, but recommend design stresses of one-third of the limiting creep stress values for other parts.

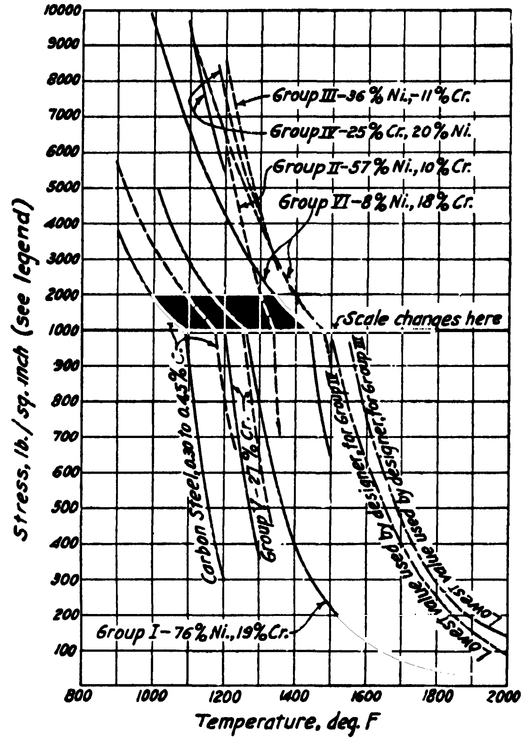

Fig. 190. Creep strength of carbon steel and nickel-chromium alloy steels, in tension.

Solid lines show stress to produce 1 per cent extension in 100,000 hr. Broken lines show stress to produce 1 per cent extension in 10,000 hr. (Compiled from tests of French, Kanter, Lobley, Norton. See Symposium on Effect of Temperature on Metals, ASTM-ASME, 1931. Data for 18-8 alloy from U. S. Steel Corporation, Allegheny Steel Company, and Associated Alloy Steel Company.)

The recommended design stress varies with the percentage of carbon, nickel, and chromium in the alloy. It is interesting to note that the carbon content of cast alloys goes up as high as 0.5 per cent. This is noteworthy because alloys of this type cannot be rolled or forged if the carbon content exceeds 0.12 per cent. In cast materials, the carbides are precipitated at the grain boundaries and offer decided resistance to slippage. It is not advisable to go beyond 0.5 per cent

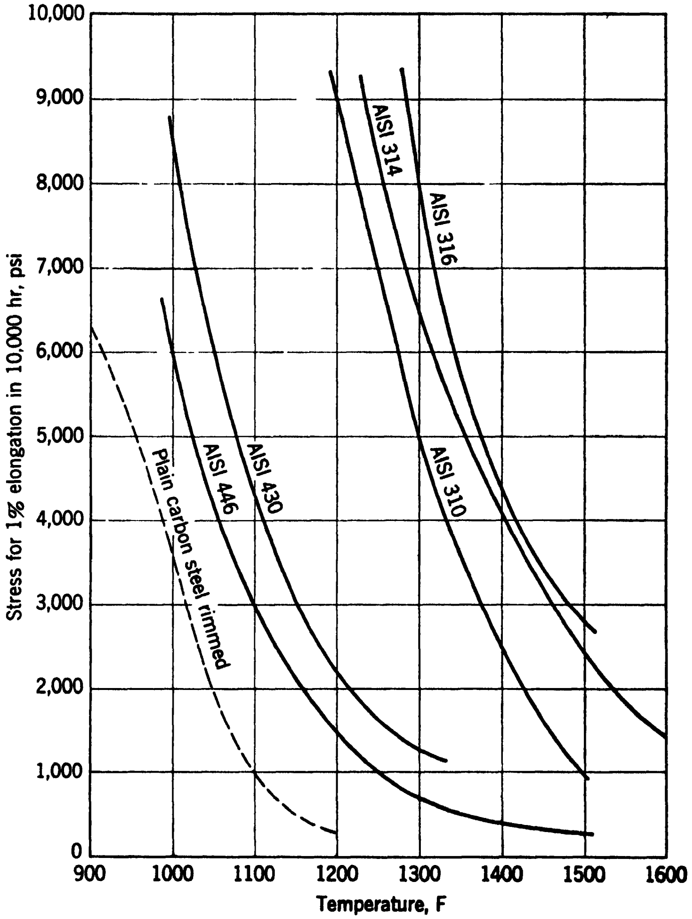

Fig. 190a. Creep strength of plain carbon and alloy rolled steels. (Various sources.)

Analyses of These Steels:

Alloy	Max C	Mn	Max P	Max S	Si	Ni	Cr	Mo	Max N_2
310	0.25	2.00	0.04	0.03	1.5	19–22.0	24–26.0		
314	0.25	2.00	0.04	0.03	1.5–3.0	19–22.0	23–26.0		
316	0.10	2.00	0.04	0.03	1.00	10–14.0	16–18.0	2.00–5.00	
430	0.12	1.00	0.04	0.03	1.00	0.50	14–18.0		
446	0.35	1.00	0.04	0.03	1.00	0.50	23–27.0		0.025

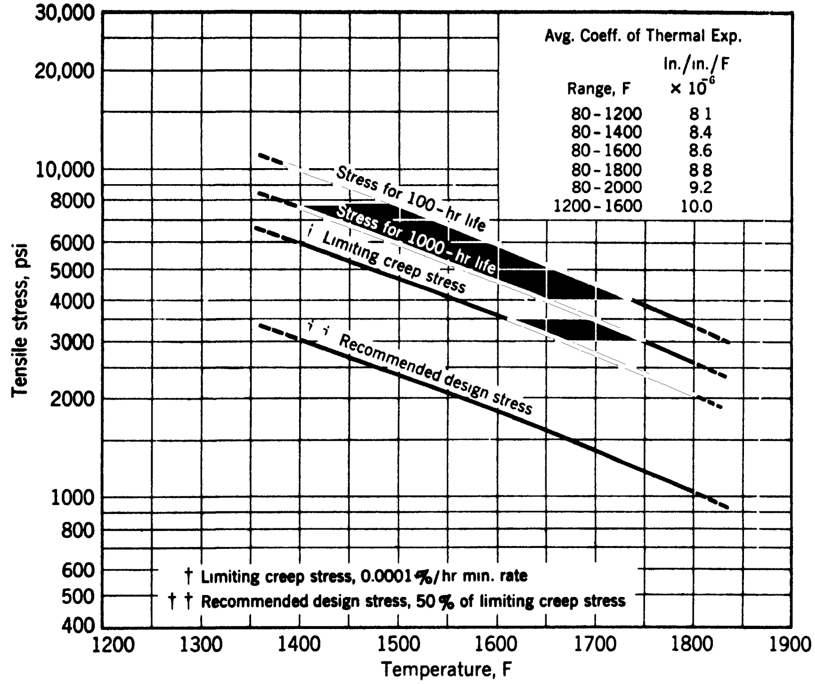

Fig. 191. Strength-temperature chart for 12 per cent Cr alloy. Elevated temperature properties, 12 per cent Cr, 60 per cent Ni, heat-resistant alloy, austenitic grade at 0.50 per cent C.

Typical Tensile Properties (Room Temperature)

Heat Treatment, F, Hr Cooling	Yield Strength, 0.2% psi	Ultimate Tensile Strength, psi	Elongation, 2 in., %	Reduction Area, %	Hardness, BHN
As cast	30,000 to 37,000	59,000 to 73,000	3 to 9	4 to 9	166 to 198
1400–24, furnace cooling	49,000 to 61,000	80,000 to 88,000	3 to 9	4.3 to 10.5	196 to 220

(Courtesy of American Brake Shoe Company.)

carbon, because alloys with higher carbon content immediately become brittle in service. Such alloys have a high coefficient of expansion and a low coefficient of heat conductivity. For that reason, they are very susceptible to stresses which are caused by changes in temperature.

With so many alloys from which to make a selection, the furnace designer may become bewildered. Without doubt, the best plan is to

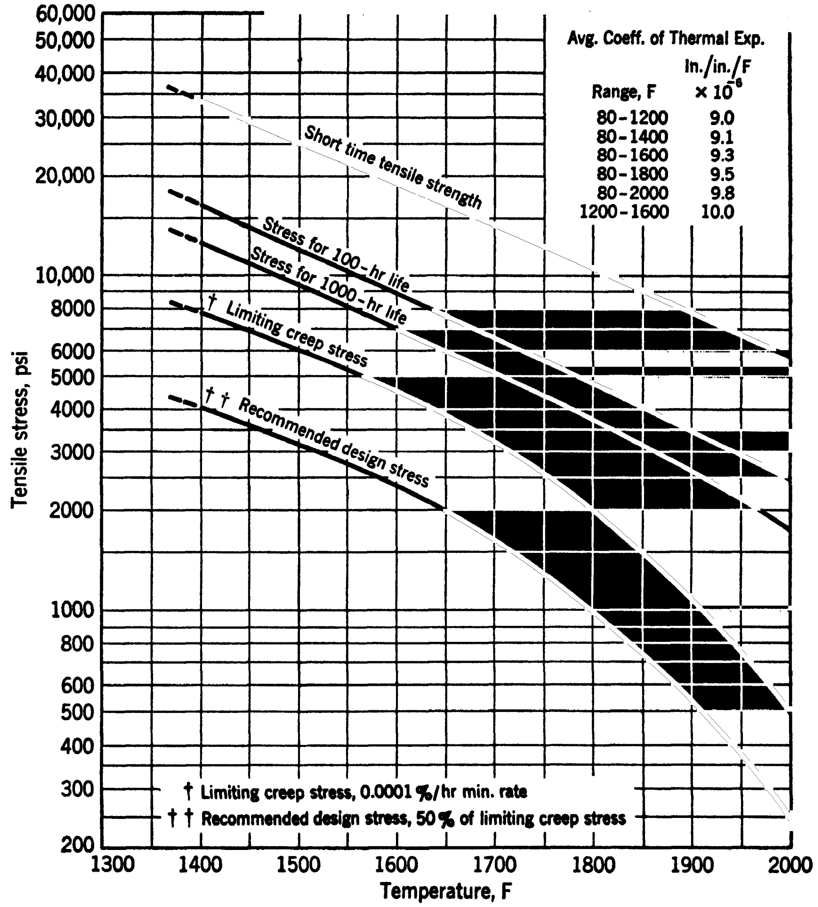

Fig. 192. Strength-temperature chart for 16 per cent Cr alloy. Elevated temperature properties, 16 per cent Cr, 35 per cent Ni, heat-resistant alloy, austenitic grade at 0.50 per cent C.

TYPICAL TENSILE PROPERTIES (ROOM TEMPERATURE)

Heat Treatment, F, Hr Cooling	Yield Strength, 0.2% psi	Ultimate Tensile Strength, psi	Elongation, 2 in., %	Reduction Area, %	Hardness, BHN
As cast	38,000 to 52,000	60,000 to 72,000	4 to 20	6 to 21	156 to 200
1400–24, furnace cooling	50,000 to 67,000	84,000 to 92,000	3 to 10	4 to 18	200 to 215

(Courtesy of American Brake Shoe Company.)

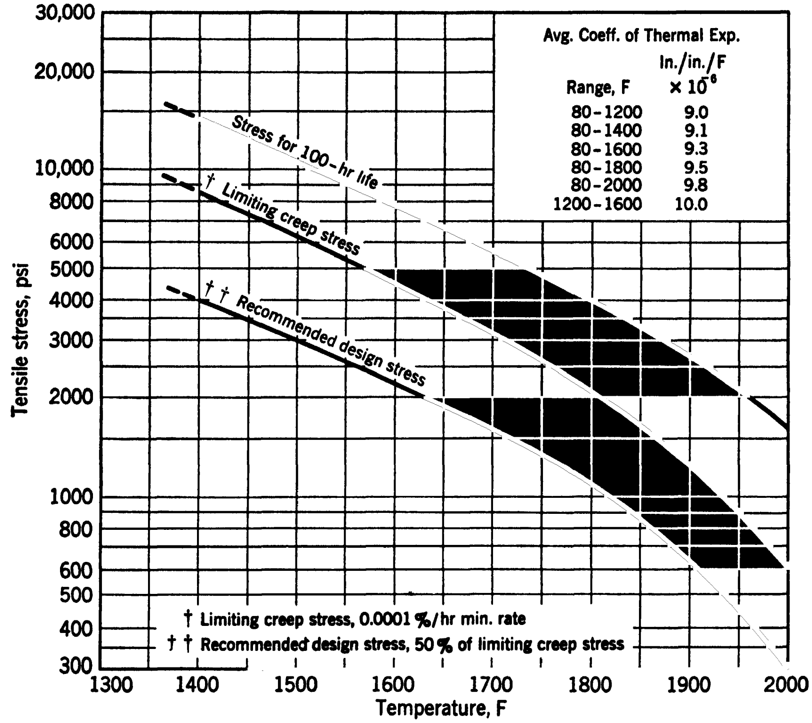

Fig. 193. Strength-temperature chart for 18 per cent Cr alloy. Elevated temperature properties, 18 per cent Cr, 38 per cent Ni, heat-resistant alloy, austenitic grade at 0.50 per cent C.

Typical Tensile Properties (Room Temperature)

Heat Treatment, F, Hr Cooling	Yield Strength, 0.2% psi	Ultimate Tensile Strength, psi	Elongation, 2 in., %	Reduction Area, %	Hardness, BHN
As cast	25,000 to 40,000	67,000 to 72,000	4 to 16	3 to 17	160 to 200
1400–24, furnace cooling	40,000 to 45,000	73,000 to 80,000	3 to 9	4 to 9	200 to 215

(Courtesy of American Brake Shoe Company.)

submit each typical furnace problem to a reliable maker of heat-resisting alloys. Such makers stress the importance of knowing the type of alloy to use. It must, however, be remembered that an alloy which was recommended for one condition may fail if subjected to a higher temperature or to a more corrosive furnace atmosphere.

In view of the somewhat complicated alloy situation, a few hints

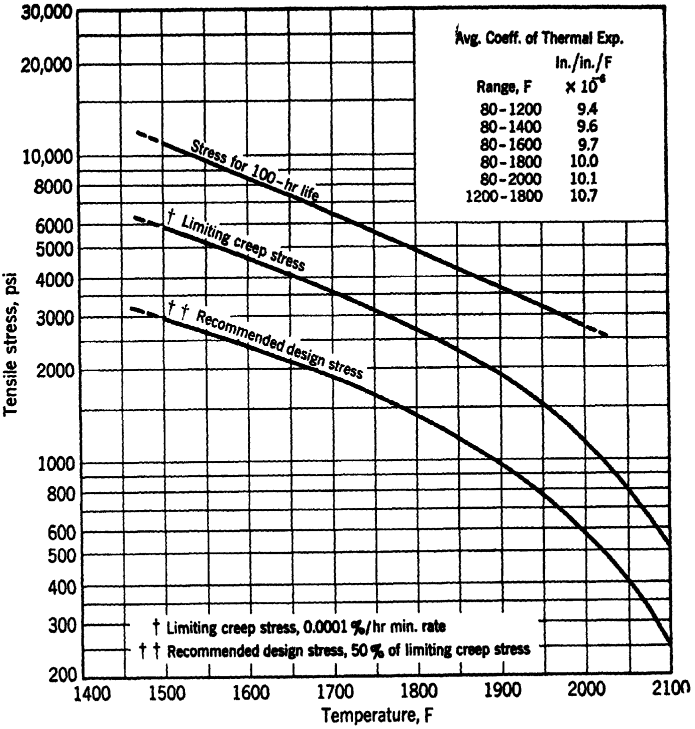

Fig. 194. Strength-temperature chart for 26 per cent Cr alloy. Elevated temperature properties, 26 per cent Cr, 20 per cent Ni, heat-resistant alloy, austenitic grade at 0.40 per cent C.

TYPICAL TENSILE PROPERTIES (ROOM TEMPERATURE)

Heat Treatment, F, Hr Cooling	Yield Strength, 0.2% psi	Ultimate Tensile Strength, psi	Elongation, 2 in., %	Reduction Area, %	Hardness, BHN
As cast	40,000 to 72,000	72,000 to 84,000	14 to 24	15 to 31	156 to 179
1400–24, furnace cooling	40,000 to 74,000	76,000 to 95,000	6 to 18	6 to 22	174 to 207

(Courtesy of American Brake Shoe Company.)

on their proper use are in order. When sulfur-bearing oil or coke-oven gas serves as fuel, alloys should have at least 24 per cent chromium, if exposed to a temperature above 1800 F. An alloy containing 26 per cent chromium and 12 per cent nickel is a favorite for this con-

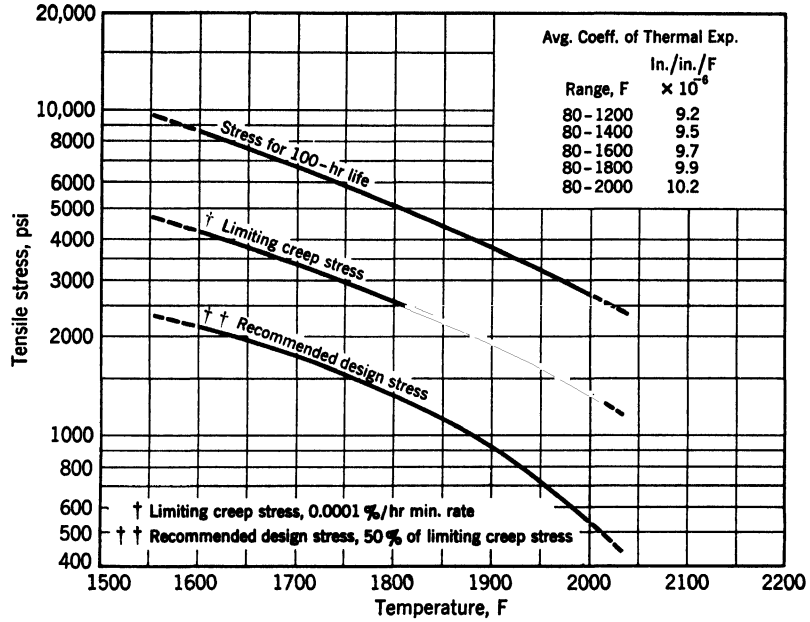

Fig. 195. Strength-temperature chart for 26 per cent Cr plus high-nickel-content alloy. Elevated temperature properties, 26 per cent Cr, 35 per cent Ni, heat-resistant alloy, austenitic grade at 0.35 per cent C.

TYPICAL TENSILE PROPERTIES (ROOM TEMPERATURE)

Heat Treatment, F, Hr Cooling	Yield Strength, 0.2% psi	Ultimate Tensile Strength, psi	Elongation, 2 in., %	Reduction Area, %	Hardness, BHN
As cast	32,000	68,000	19	18	170
1400–24, furnace cooling	46,000	78,000	6	5	175

(Courtesy of American Brake Shoe Company.)

dition. In the absence of sulfur, this alloy can be strengthened by raising the nickel content to 15 or even 20 per cent.

When sulfur conditions are extremely severe and the stress is comparatively low, an alloy containing 29 per cent chromium and 9 per cent nickel serves well. It has great resistance to sulfur attack, at the expense of strength.

When the sulfur problem is absent, an alloy containing 35 per cent nickel and 15 per cent chromium is widely employed. Most of the miscellaneous furnace parts used in automotive and aircraft plants are made from this alloy. It has good strength and excellent resistance to

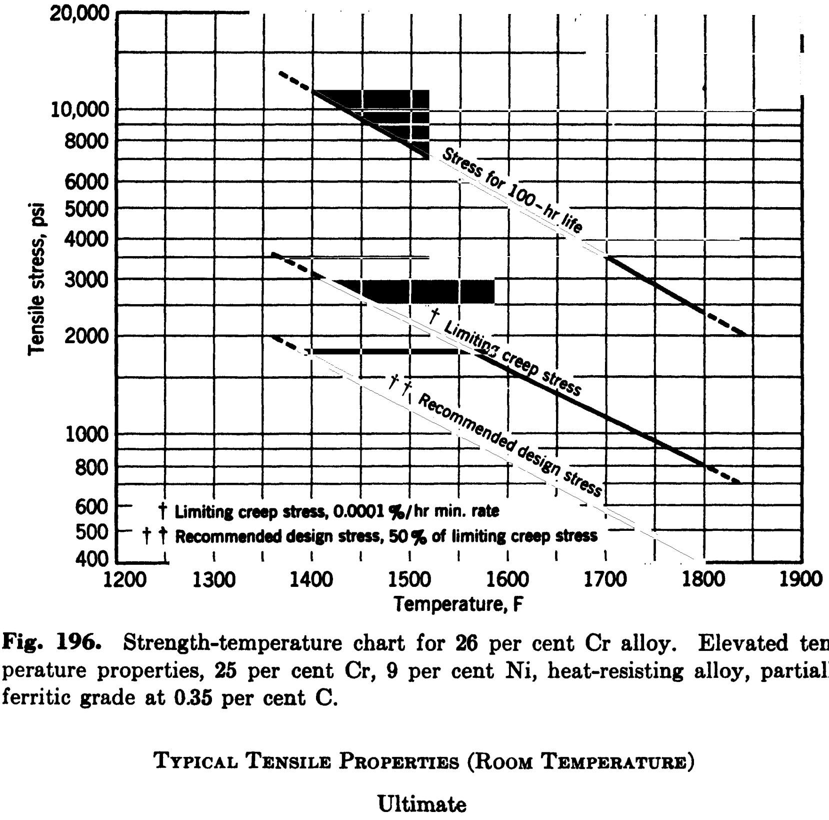

Fig. 196. Strength-temperature chart for 26 per cent Cr alloy. Elevated temperature properties, 25 per cent Cr, 9 per cent Ni, heat-resisting alloy, partially ferritic grade at 0.35 per cent C.

TYPICAL TENSILE PROPERTIES (ROOM TEMPERATURE)

Heat Treatment, F, Hr Cooling	Yield Strength, psi	Ultimate Strength, psi	Elongation, 2 in., %	Reduction Area, %	Hardness, BHN
As cast	42,000 to 76,000	95,000 to 107,000	4.5 to 18.0	2.7 to 17.0	207 to 269
1400–24, furnace cooling	60,000 to 70,000	79,000 to 88,000	0.0 to 4.5	0.0 to 6.2	226 to 387

(Courtesy of American Brake Shoe Company.)

oxidation at temperatures up to 2000 F. Since this alloy has considerable resistance to thermal fatigue, it is normally utilized for large retorts, calcining tubes, and other sizable castings involving thermal gradients, as well as for moving parts and those subjected to bending stresses. The chromium-predominating alloys are ordinarily used for skids, hearth plates, and other parts which do not move and are suitably supported in the furnace. When temperature changes are extremely severe, alloys containing either 60 per cent nickel and 12 per cent chromium or a special alloy such as that of group III in Table 19

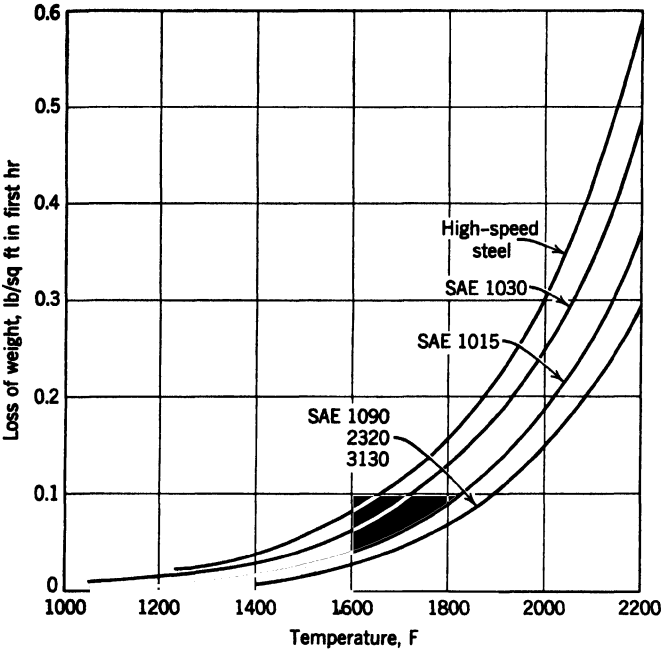

Fig. 197. Comparison of the rate of scaling of various steels at elevated temperatures.

Analyses of these steels:

			Max	Max		Max	Max	Max		
SAE	C	Mn	P	S	Si	Ni	Cr	Mo	Va	W
1015	0.13 to 0.18	0.30 to 0.60	0.04	0.05	0.15 to 0.30	0.25	0.20	0.06		
1030	0.28 to 0.34	0.60 to 0.90	0.04	0.05	0.20 to 0.35	0.25	0.20	0.06		
1090	0.85 to 0.98	0.60 to 0.90	0.04	0.05	0.20 to 0.35	0.25	0.20	0.06		
2320	0.18 to 0.23	0.40 to 0.60	0.04	0.04	0.20 to 0.35	3.25 to 3.75	0.20	0.06		
3130	0.28 to 0.33	0.60 to 0.80	0.04	0.04	0.20 to 0.35	1.10 to 1.40	0.55 to 0.75	0.06	1.00 to 1.25	17.5 to 18.5
High speed	0.71 to 0.75	0.20 to 0.40	0.03	0.03	0.20 to 0.40	0.50	3.75 to 4.25			

are recommended (in the absence of sulfur). All of these alloys, as already mentioned, can be modified for specific service by altering the carbon content, upward for strength, downward for ductility.

One of the most corrosive elements is oxygen. Oxidation of iron and steel becomes serious at about 1000 F and increases by leaps and bounds as the temperature is increased. Figure 197 gives loss of weight in pounds per (sq ft, hr) for several steels in a furnace atmosphere resulting from perfect combustion of manufactured gas, based upon tests reported by Murphy and Jominy, *Research Bulletin* 21, October 1931, University of Michigan.

Mild steel and alloy steel are the materials for many muffles, such as hoods in annealing furnaces for sheets and coils of strip. The furnace engineer is often confronted with the question, whether frequently replaced muffles of plain carbon steel or more durable muffles of alloy steel are the more economical. The decision is affected by the relative corrosion and by the price of alloys. This price varies with market conditions and cannot be discussed here, but scale formation can be discussed.

Metallic surfaces of the nickel-chromium alloys oxidize readily, but a thin, closely adhering film of oxides of nickel and of chromium forms on the surface and resists oxygen penetration; thereafter the film increases in thickness very slowly. With some of the alloys, if the oxide film is first formed at moderate temperatures, it is more adherent and less likely to flake off in high-temperature service than if initially formed at the high temperatures. As previously mentioned, the presence of sulfur compounds usually causes rapid corrosion (pitting) of alloys containing nickel.

In so-called reducing atmospheres, scaling is decreased, but by no means absent.

In an effort to raise the allowable working temperatures of electrical resistor furnaces, a number of metals and alloys have been developed for the heating elements (for details see *Industrial Furnaces,* Volume II).

Many attempts have been made to extend the usefulness of unalloyed steel in the region of high temperatures by protective coatings. One of the best methods bears the name "calorizing" (called "aletizing" in Europe). "Calorized metal" is cast iron or soft steel, the surface of which is protected by alloying with aluminum, producing an aluminum oxide film. This protects the iron from oxidation up to 1750 F.

By a special process, the aluminum-iron alloy film has been increased to about 0.020- to 0.030-in. thickness and the brittleness is much reduced. The cost of calorizing is rather high. It is frequently cheaper to use iron or steel without protection and to renew the pieces from time to time.

Refractories. The furnace proper (walls, roof, hearth) must be constructed of materials which remain rigid and strong at furnace temperatures and which are poor conductors of heat. Another requirement is that furnace atmospheres and slag affect them very little. Such materials are called refractories. With but few exceptions, refractory materials are oxides. Table 20 furnishes information on refractory materials in their commercial (i.e., impure) condition.

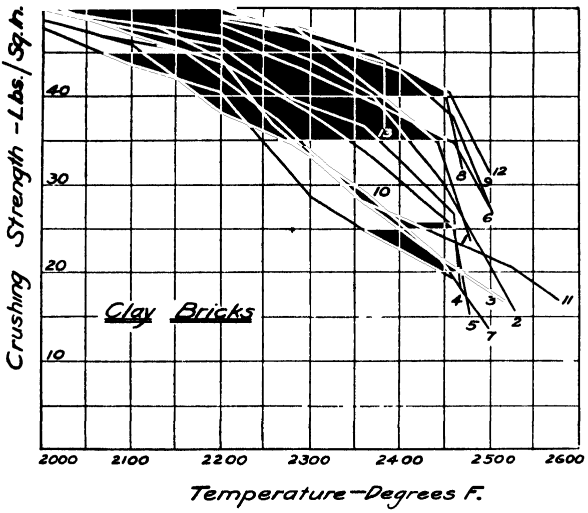

Fig. 198. Crushing strength as a function of temperature for various standard brands of fireclay bricks.

The common rule is that the cheapest refractory material that stands up under the prevailing conditions (of temperature, load, furnace atmosphere, and slag) is selected. For the great majority of industrial furnaces that material is fireclay. The term "fireclay" includes materials of many different compositions. And the term "fireclay brick" or, for short, "firebrick" includes, in addition to differences in composition, bricks produced by different methods of manufacture. In consequence, firebricks have widely varying properties, as illustrated by Fig. 198 which shows clearly that the relation between temperature and crushing strength is far from uniform for firebricks of different brands. The tests from which Fig. 198 was plotted were made more than thirty years ago. Today the differences are even greater, because the quality of firebricks is now superior to that which is known as dense firebrick. Lightweight firebricks, also known

TABLE 20. Representative Chemical Analysis of Refractories*

	Per cent of											
		Iron Oxides										
	Silica, SiO_2	Fe_2O_3	FeO	Alumina, Al_2O_3	Lime, CaO	Magnesia, MgO	Soda, Na_2O	Potash, K_2O	Water, H_2O	Titania, TiO_2	Chromic Oxide, Cr_2O_3	Manganese Oxide, MnO
Low-grade fireclay	60.5	2.35	0.40	24.90	0.25	0.05	0.15	...	10.0	1.40		
High-grade fireclay	50.35	0.75	...	33.65	0.10	0.05	0.10	0.40	13.75	0.80		
Low-grade fireclay brick	61.72	6.43	...	28.6	0.46	1.04	0.05	0.05	...	1.60		
High-grade fireclay brick	53.52	2.00	...	41.0	0.30	0.30	0.90	0.20	...	1.60		
Kaolin brick	51.96	0.57		45.38	0.10	0.22	...	...	...	1.73		
High-alumina brick, 50%	41.8	1.8	...	51.8	0.1	0.5	1.3		...	2.7		
High-alumina brick, 60% (sillimanite brick)	35.9	0.5	...	61.2	0.4	0.1	0.5		...	1.4		
High-alumina brick, 70% (mullite brick)	22.7	1.8	...	70.8	0.1	0.3	0.9		...	3.7		
High-alumina brick, 80%	15.7	1.8	...	79.0	0.2	0.3	1.0		...	2.2		
Silica brick	96.42	0.50	...	0.75	2.01	0.08	...	...	...	0.06		
Dead burned magnesite	3.96	5.81	...	1.95	3.0	84.85						
Chromite	5.5	...	13.50	17.00	Trace	21.7	...	...	...	...	41.5	0.80

---

* Adapted from Camp and Francis, and modified according to information received from makers of refractory material. Where totals do not add to 100 per cent, traces of other compounds are present.

insulating refractories, although made of the same materials, are less strong because the bricks are full of pores.

The differences are, at least in part, explained by the phase diagram of silica-alumina (see Fig. 199) which, by its upper curve, furnishes

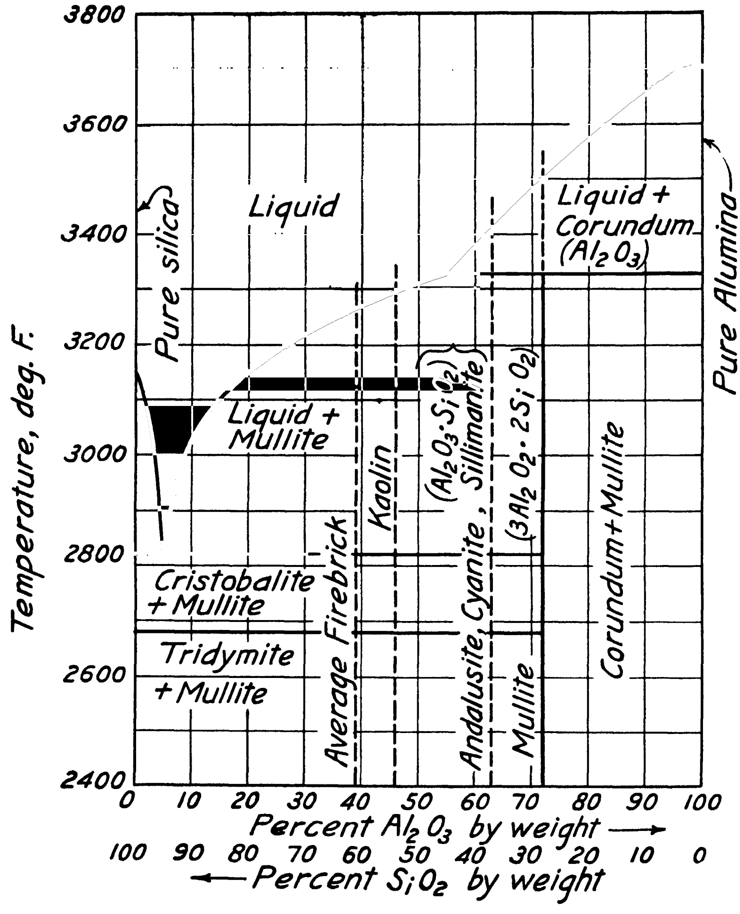

Fig. 199. Fusion temperatures of silica-alumina mixtures. (Chiefly from Sosman, *Journal American Ceramic Society,* 1933, p. 65.)

the relation between silica-alumina ratio (abscissae) and fusion temperature of the mixture (ordinates). For the composition 3 per cent alumina and 97 per cent silica, the fusion temperature is about 3030 F. For the composition 29 per cent alumina and 71 per cent silica, which corresponds to a low-grade firebrick, the apparent fusion temperature lies above 3200 F. This temperature is higher than the fusion temperature of pure silica, and yet at 3000 F a silica brick steadfastly bears loads under which the low-grade firebrick is gradually but unceasingly squashed. The same is true for an average firebrick which, according to Fig. 199, fuses at a temperature slightly below 3300 F. The reason for this anomalous behavior is the presence (in the brick) of the eutectic, which melts at 2820 F. The presence of this glassy eutectic causes the bricks to become soft before they melt.

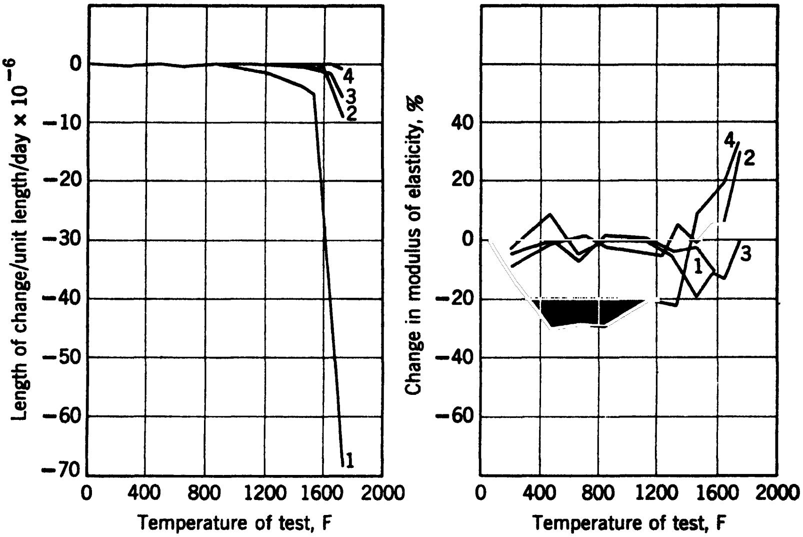

Fig. 200. Change in length and modulus of elasticity with temperature, for bricks in compression.

1 and 2 Firebricks
3 Alumina brick
4 Silica brick

However, this is not the whole story. From Table 20 it can be seen that fireclay contains iron oxide. This latter constituent forms a glassy eutectic with the silica of fireclay and depresses the fusion point to 2200 F. Low-grade fireclay contains additional fluxing material which depresses the softening point still more and makes the firebricks worthless for high temperatures. High-grade firebricks are produced from clays that contain very little iron oxide and other fluxes.

The glassy eutectics in refractories are viscous. They yield at high temperatures and when subjected to stress. In consequence, low-grade firebricks change their shape at high temperatures and if loaded. Figure 200* illustrates their behavior, if the refractories are compressed. Figure 201* refers to yielding if the refractories are in tension. Refractories are now subjected to tension, for instance, in suspended roofs and in chain-walls. In such applications tensile stress is often increased by bending action. For that reason, furnace engineers usually consider the tensile strength of refractories to be only one-third of the compressive strength. Another factor enters: Refractories that are subjected to tension have odd shapes in which shrinkage cracks are apt to occur. In compression, a crack transmits force; but in tension, the crack not only transmits no force, but it also acts as a stress raiser. At the edge of the crack, the stress often is nine times as high as it would be without the crack.

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* The curves of Figs. 200 and 201 were adapted from a greater number of curves published by L. E. Mong in the *Journal of American Ceramic Society,*

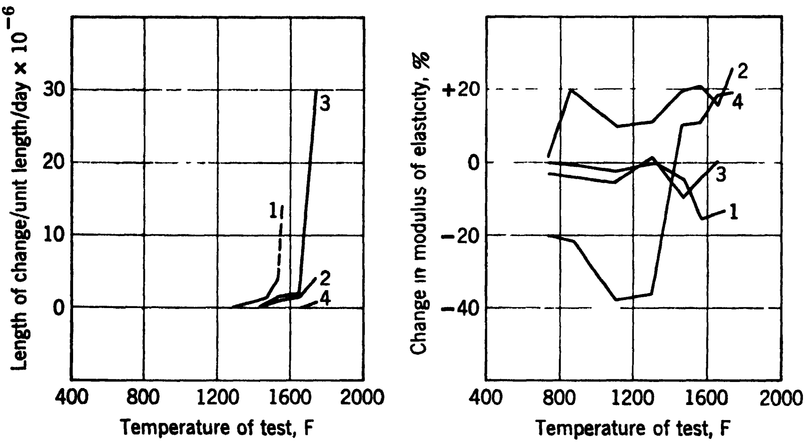

Fig. 201. Change in length and modulus of elasticity with temperature, for bricks in tension.

1 and 2 Firebricks
3 Alumina brick
4 Silica brick

Ascertaining the tensile strength of refractories at elevated temperatures is difficult and costly. For that reason, an approach to the tensile strength is obtained by testing refractory bricks in bending. The stress at which they fail is computed from the standard beam equation and is called the modulus of rupture. Since, in bending, the inner portions of the brick support the skin, the modulus of rupture is greater than the tensile strength. The latter is roughly two-thirds of the modulus of rupture.

In spite of the great number of variables which affect the strength of firebricks, selection of the most suitable type is usually not difficult.

For temperatures not exceeding 1400 F, low-duty firebricks suffice. Their fusion point is 2770 F, which corresponds to pyrometric cone-equivalent PCE 19. For temperatures up to 1700 F, "moderate-duty" firebricks are suitable. They range between 26 and 28 PCE; this is equivalent to the range between 2903 and 2939 F. For furnace temperatures (which are always higher than the temperature of the charge) up to 2200 F, "intermediate-heat-duty" bricks are advisable except in tension and in places where the high temperature can penetrate deeply into the brickwork. Such bricks have a PCE of 28 to 31, which is equivalent to the range between 2939 and 3056 F. Temperatures up to 2500 F are encountered in modern furnaces. For such temperatures "high-duty" bricks, usually aluminous, serve very well. They are rated at 31 to 33 PCE, that is to say, between 3056 and 3173 F. Temperatures above 2500 F are encountered in "high-speed" heating furnaces, and "superduty" bricks are available with PCE 33 to 34, in the temperature range between 3175 and 3200 F. The fusion temperatures lie at a much higher level than the furnace temperatures. The reason, as may be remembered, lies in the fact that firebricks become soft and yield under load at temperatures that lie several hundred degrees Fahrenheit below their fusion points.

Strength and durability of firebricks are affected not only by temperature, but also by other factors. Among these are (1) temperature gradient in bricks, and (2) superimposed load. Temperature gradient has two effects, one of which is illustrated by Fig. 202. If a thick or well-insulated wall of low-grade firebrick is exposed to high temperature, the soft, mushy layer of brick on the hot face is so thick and heavy that it gradually sloughs off. In a thin wall, the soft layer is so thin that it sticks. If it sloughs, the process is extremely slow. This situation has led to an absurd practice. Roofs and walls are often left without insulation in the regions of highest temperature, where insulation would save the greatest amount of fuel. This practice is a leftover from the days when superduty bricks were not available.

Although a steep temperature gradient prevents sloughing, it produces an undesirable effect, namely, spalling. Like all other materials, refractories expand with rising temperature. From Fig. 203 it can be seen that the expansion characteristics of refractories differ, and also that, for each refractory, a temperature range with rapid expansion exists. If, within a short distance, a great difference in temperature exists within a refractory, spalling occurs unless the great temperature difference comes within the plastic range of the refractory. Severe temperature gradients are produced transiently by rapid heating, or permanently if a thin wall is water-cooled on one side while the other side is exposed to furnace temperature.

The rapid expansion of silica brick in the brittle range below 1200 F is the reason why the use of such bricks is avoided in industrial furnaces that are shut down overnight or over the week end. Furnaces built of silica bricks must be heated up very slowly and preferably

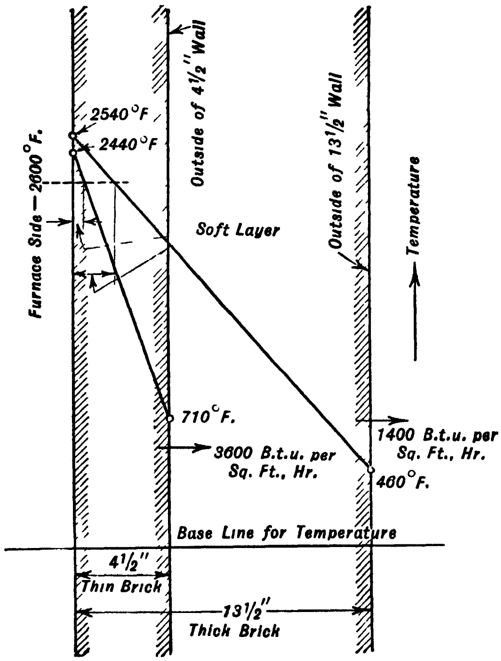

Fig. 202. Temperature gradient through furnace walls of different thickness (equal pressure on both sides of the wall).

kept above 1100 F during each shutdown. Fireclay bricks ordinarily do not spall unless the heating up process is carried out at an excessive rate.

A thin wall of firebrick is no protection against heat loss to water-cooled parts. The shear stresses are so great that the wall is rapidly disintegrated and disappears.

The next remark applies not only to firebricks but to all refractory materials. No refractory material can withstand harsh mechanical treatment. Severe bumping by the crane or charging machine loosens the bond between bricks and initiates destruction. Lighting a furnace that contains a gas-and-air mixture irreparably damages the furnace structure.

Firebrick in the form of special shapes is usually not as strong as it is in standard shapes. The thicker the sections, the greater is the danger of internal stresses and of cracking in service. And special shapes which contain both thick and thin sections are expensive, for the reason that the manufacturer of refractories sometimes must make 25 such tiles in order to obtain one good one. Whenever possible, standard shapes are preferable. The catalogues and price lists of manufacturers of refractory materials show pictures and dimensions of the regularly available standard shapes. In passing, it may be remarked that two series of standard bricks are now available, one of which is based on 2½-in. thickness of brick, while the other one is based on 3-in. thickness. The 3-in. series is gaining in favor, for two reasons: (1) A bricklayer can lay just as many 3-in. bricks in one hour as 2½-in. bricks; (2) There are fewer joints and, in consequence,

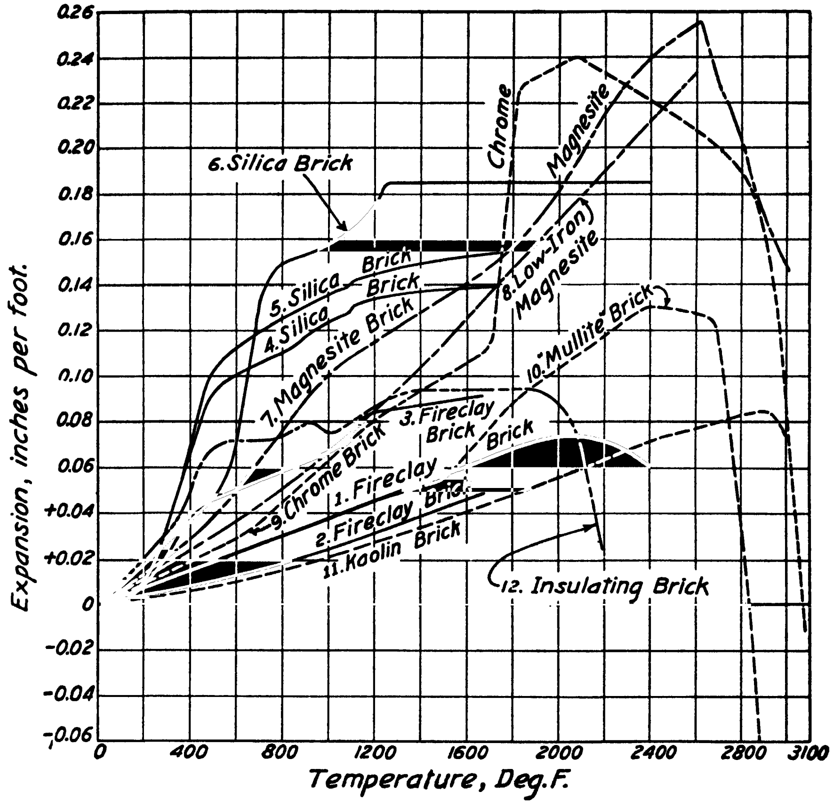

Fig. 203. Thermal expansion of refractories.

fewer exposed edges and corners to be attacked by furnace gases. The use of firebricks in furnace hearths is discussed in the section on the strength of hearths.

For the sake of completeness, two refractory materials which were mentioned by name in Fig. 199 must be discussed. They are (1) Al_2O_3—SiO_2, which is known as sillimanite, kyanite, or andalusite, and (2) $3Al_2O_3$—$2SiO_2$. This material is known as mullite, because it was found on the island of Mull.

Both of these materials are excellent refractories, especially if they are free from impurities. The only safe way to produce them in the pure state is to mix pure alumina and pure silica in the proper proportions and to melt the mixture in an electric furnace. High-alumina materials produced in this manner are among the best existing refractories. Mullite consists of needle-shaped crystals which interlock, thereby producing great strength and resistance to spalling. In industrial furnaces, mullite bricks are used only occasionally. These bricks resist spalling and flame impingement in hot spots of forge furnaces. They also give good service in piers (between doors), support-arches, and other load-bearing constructions in furnaces that are exposed to very high temperature.

In the preceding paragraphs spalling has been mentioned. A material that has great resistance to spalling is silicon carbide. In comparison with other refractories, it has high thermal conductivity (for which see Fig. 357) and great strength. Variations of the temperature of the surface are quickly communicated to the whole mass, and the temperatures are quickly equalized. The recrystallized silicon carbide bricks spall about twice as much as the clay-bonded type containing 3 to 10 per cent clay, but still much less than fireclay bricks. The coefficient of expansion of silicon carbide bricks is 0.0000026 per degree Fahrenheit, which is slightly below that of firebricks. They can be bonded with firebricks for temperatures that occur in heating and annealing furnaces. Fireclay mortar, fireclay refractory cement, or silicon carbide cement may be used for bonding.

Silicon carbide is well suited as material for muffles, for hearths in underfired furnaces, for hearth rails, skid bars and any other parts of a furnace in which high heat conductivity, abrasion resistance, and hot strength are advantageous. Silicon carbide is slowly attacked by basic slags, or molten iron or steel. The bulk specific gravity of silicon carbide refractories is from 2.1 to 2.6. The material is manufactured in brick form and is sold under trade names. The specific heat of these bricks is 0.180 to 0.186; the weight of recrystallized silicon carbide brick (98 to 99 per cent silicon carbide) is 8.5 to 9.0 lb, while that of a clay-bonded silicon carbide brick is 9.25 lb. The size of a brick is 9 by 4½ by 2½ in. The crushing strength of the recrystallized brick when cold is 12,500 psi; of the clay-bonded silicon carbide brick, 14,700 psi. Cross-bending tests show that at 2500 F the strength of recrystallized brick averaged 5 per cent higher than when cold; the strength of clay-bonded silicon carbide brick varies from 80 to 108 per cent of its strength when cold. No sign of failure was observed at 3000 F under compressive stress of 50 psi. Bonded silicon carbide bricks withstand the action of oxidizing flames or furnace atmospheres except at temperatures that are in excess of those occurring in industrial furnaces. The bonded material does not withstand the action of reducing atmospheres above 2800 F, but the recrystallized material is inert to reducing conditions up to the decomposition point of the silicon carbide itself. It may be mentioned that silicon carbide is a refractory which is not an oxide.

The thermal conductivity of silicon carbide refractories varies with composition and method of manufacture. At furnace temperatures, it roughly equals 100 Btu in. per (sq ft, hr, F).

A refractory material of high conductivity, which was formerly used in industrial furnaces to a much greater extent than it is now, is magnesite. In the types of furnaces discussed in this book, magnesite is not a factor in refractory construction.

The position formerly held by magnesite is now held by chrome ore and chrome bricks. Chrome ore is chemically neutral and is not attacked by iron oxide. It can be purchased in the form of powder or ramming mixture, or as chrome brick. In furnaces that are built of acid and basic refractories, chrome brick is placed between the two, in order to prevent chemical action. Chrome ore is heavier than fireclay, weighing between 220 and 230 pcf. In industrial furnaces chrome ore is almost entirely used in the form of plastic ramming mixture. The plastic is made by grinding the ore and mixing it with not more than 5 per cent of binder, which gives it air-setting as well as hot-setting properties. The plastic chrome material is molded or pounded into place with a hammer. An air-hammer is preferred, if there is room for using it. After installation, the chrome plastic has a density of approximately 200 pcf. It forms a hard, smooth surface to which (up to 2600 F) steel does not stick and which resists abrasion very well. Scale is easily removed from it without damage to the hearth. The material is sold in the United States under various trade names, such as K-N Plastic, Thermolith Batch, G-R Plastic Chrome, and

Kromepatch. Chrome bricks (seldom used in industrial furnaces) weigh 180 to 195 pcf. The difference is caused by impurities; commercial chromite contains MgO, Al_2O_3, and SiO_2.

Silica firestone, known in Germany as Quartz Schist or Mica Schist, is a natural product which was formed millions of years ago by the cementing of grains of silica by interstitial matter under great pressure. It consists of about 93 per cent silica and 4 per cent alumina, and has a fusion point of about 3000 F. It is produced to specification in size and shape, and is widely used in large blocks for soaking-pit linings which operate at constant temperature, for ladle linings, and for other applications where temperatures are high and abrasion is severe.

At the other end of the silica-alumina series is fused alumina, which has high thermal conductivity, great strength at furnace temperatures, and is not affected by contact with metals. It was extensively used for suspending metallic resistors, but has largely been replaced by metallic hooks.

Zirconia is used in small quantities in burners that have to withstand heat approaching adiabatic flame temperatures.

Castable Refractories. Since about 1945, many hearths, walls, and roofs have been constructed of castable refractory concrete, which is monolithic. The cementing material is calcium aluminate which, if pure, withstands temperatures slightly higher than 3200 F. When mixed with refractory aggregate, it becomes refractory concrete. The base material (calcium aluminate) is sold in the United States under the trade name "Lumnite." The composition of the refractory aggregate must be adapted to the furnace temperature to which it will be exposed. Aggregate in sizes between ¾ in. and 1½ in. is preferred. One volume of cement is mixed with 6 to 7 volumes of aggregate. For sealing, less aggregate and smaller sizes are required. Refractory concrete may either be cast in place or precast.

Upon being heated, refractory cement shrinks. By the selection of a suitable aggregate, a refractory concrete can be produced that does not change in volume up to 2000 F. At higher temperatures, linear shrinkage does not exceed 2 per cent. In a few instances structures of refractory concrete have been damaged by cracking, which can be minimized if the material is purchased from an experienced maker and if temperature conditions are clearly specified.

After drying at 230 F, the density of refractory concrete lies between 120 and 150 lb per cu ft, varying with the density of the aggregate. Depending upon the composition of the aggregate and upon the temperatures to which the concrete has been heated, the cold crushing strength lies between 1000 and 8000 psi. If the aggregate is an insulating material, the mixture is a castable insulating concrete which is sometimes called "lightweight castable." Its density ranges between 25 and 85 lb per cu ft. Lightweight castables are useful up to 2600 F, if the aggregate is suitable. The cold crushing strength of lightweight castables lies between 100 and 2000 psi.

Plastic Refractories. Fireclay is plastic. Instead of being molded into bricks and burned in a kiln, fireclay may be rammed into place in an industrial furnace and burned by the heat of the furnace, if the temperature of that furnace is high enough. The method of burning fireclay "in situ" is old in boiler (steam generator) practice, but is comparatively new in the construction of industrial furnaces. The plastic fireclay exhibits different properties, depending upon its purity. The plastic is made by various producers in the United States. The properties of the burned refractory are also affected by the relative amounts of water that is added before the plastic is rammed into place.

Surplus water reduces the ultimate strength, but results in an insulating refractory, because the finely subdivided water evaporates, leaving small cavities. Slow heating is a prerequisite for obtaining perfect walls.

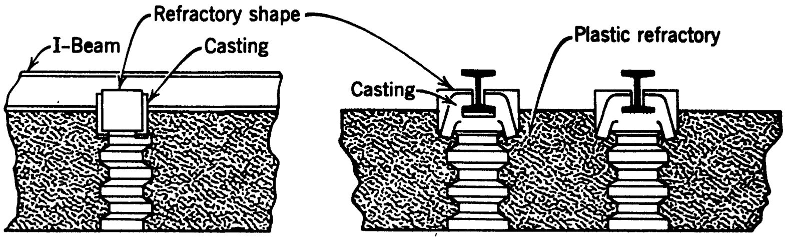

Fig. 204. Arrangement of anchors for plastic refractory.

The monolithic mass (rammed plastic clay) is held in place by anchors that are attached to the binding. The anchors may be metallic or ceramic. Similar anchors are used for castables. Arrangement and use of anchors is shown in Fig. 204 which represents a portion of a suspended roof. The construction for vertical walls is similar. In furnace parts that are subjected to frequent and severe changes of temperature (for instance, the covers of soaking pits), rammed-plastic firebrick lasts longer than castables.

In the use of castables and of plastic fireclay, the makers of these materials should be consulted.

Insulation. By this term is meant a material that has a much lower thermal conductivity than dense firebricks. Insulating firebricks have that property but are commonly not called insulation. Insulation is not exposed to the interior of furnaces, because it collapses, melts, or otherwise deteriorates at furnace temperatures.

From earlier chapters, it may be judged that good insulators contain a large number of very small air spaces. A consequence is that good insulators have but little mechanical strength. Among such insulators are silocel, magnesia, fibrous materials such as asbestos, mineral wool, slag wool, a silicious foam, and still others. Silocel is a natural, soft stone which can be sawed and cut easily. It is used either raw or calcined. Silocel is also available in the form of powder and is then called diatomaceous earth.

At the time of this writing, fibrous insulation is given preference over granular (powdery) insulation. The fibers are either asbestos or similar material. This insulation is known as block insulation or as blanket insulation, depending on thickness and extent. The thermal conductivity of fibrous insulation is lower than that of insulating refractory given in Table 21. In spite of its low density (20 to 22 pcf) the blocks and blankets have good mechanical strength. In hood-type furnaces blocks have been exposed to hot face temperatures not exceeding 1400 F. In the United States this type of insulation is sold under the trade name "Superex." Other brands of almost identical properties are available.

A very light and yet strong insulating material is sold under the trade name "Foamsil."

Slag wool and mineral wool are mixed with a binder, such as lime mortar and molasses, before being applied.

By selecting the proper aggregate, refractory concrete can be made to serve as insulation.

The makers of insulating materials furnish information on the highest safe temperature and the highest safe stress to which various insulating materials may be subjected.

Insulating Refractories. Firebricks that contain many small pores are known as lightweight firebricks or as insulating refractories. The basic material is fireclay. The properties of lightweight bricks are more varied than the properties of dense firebricks because, in addition to the properties of the basic material, the size, number, and arrangement of the pores affect the properties. Lightweight firebricks are available for hot face temperatures from 1600 (or below) up to 3000 F. The bricks that are made for the highest temperatures consist of the best material and have the smallest number of pores. The bricks that are intended for lower temperatures are more porous and are, for that reason, better insulators. The formation of the closed pores in the bricks is controlled by the method of manufacture. The standard

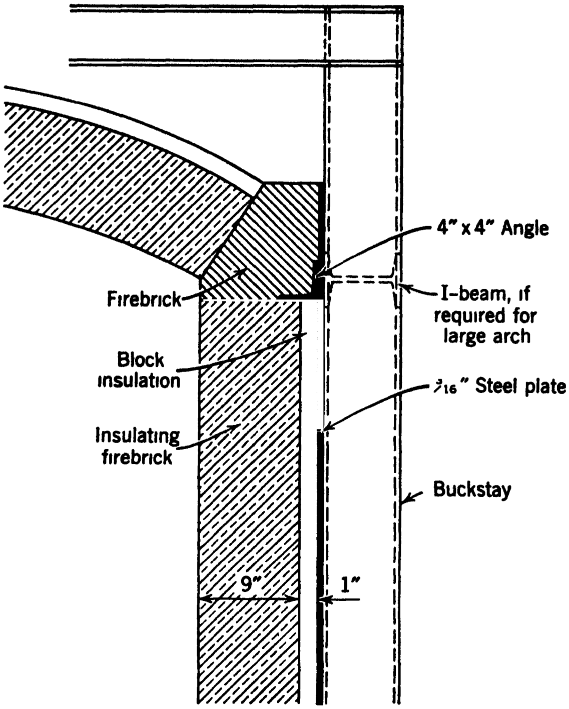

Fig. 205. Typical arrangement of furnace wall.

groups of such bricks, including some of their principal properties, are given in Table 21.

TABLE 21

Group	16	20	23	26	28	30
Maximum temp, F, exposed	1600	2000	2300	2600	2800	3000
Weight, pcf	19–29	25–35	26–42	42–48	43–58	50–65
Fusion point, F	2700	2750	2750	3000	3190	3190
Cold crushing strength, psi	48–70	95–115	117–170	154–190	200–250	300–400
Approximate thermal conductivity, Btu in./(sq ft, F, hr)						
at 1000 F	1.0	1.4	1.7	2.0	2.2	2.9
at 2000 F	...	...	2.1	2.8	3.0	3.8

Insulating refractories, because of their light weight (which is of great advantage in movable hood-type furnaces) and because of their great saving in heat storage, have become almost standard material of construction in industrial heating furnaces (Fig. 205 shows a typical construction). Exceptions to this statement include furnace hearths, piers and walls around doors (door jambs) where abrasion would wear these bricks away at an excessive rate. By control of the composition of the basic material, the bricks can be made to resist highly reducing atmospheres, and are almost exclusively used in the lining of atmosphere-gas generators and in atmosphere furnaces where such gases are used to protect the charge in the furnace.

2. THE STRENGTH OF FURNACE ROOFS*

After this enumeration of the properties of the materials that enter into furnace construction, it is desirable to consider the places where failure is most likely to occur. Prominent among such places are roofs (see Fig. 206). Whenever material spans an opening, there must be tension in some part of the covering. Because hot refractories are extremely weak in tension, it is usually taken up by cold steel lying outside of the furnace. The refractory part of a roof can then be put wholly into compression.

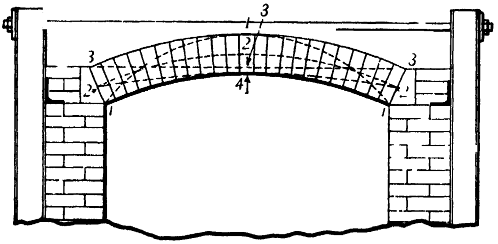

Fig. 206. Possible lines of thrust in an arch.

This idea is carried out in the commonest type of roof, which is the arch or vault. Though simple in construction, the arch is complicated in theory because it is, in the language of mechanics, three-fold statically indeterminate—which means that we cannot gain any knowledge of the stress distribution in the arch unless we investigate the elastic and plastic deformations of the arch proper, and of its abutments. In a cold arch, the centerline of force may follow any curve that is compatible with stability; and that curve will depend on the method of bedding. In Fig. 206, any of the dotted lines 1–1–1, 2–2–2, or 3–3–3 is possible, as well as many others. Since an arch is only an inverted equilibrium polygen (the catenary is the ideal shape), the internal forces are smallest if they follow the line 1–1–1 and are greatest if they follow the line 3–3–3. If the latter were straight, the force would have to be infinitely great; but, of course, that condition cannot occur because the terrific compression at the affected points would reduce the brick thickness, and thereby automatically transfer the stress to adjacent brick sections. This would curve the line of force.

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* See also "sidewalls."

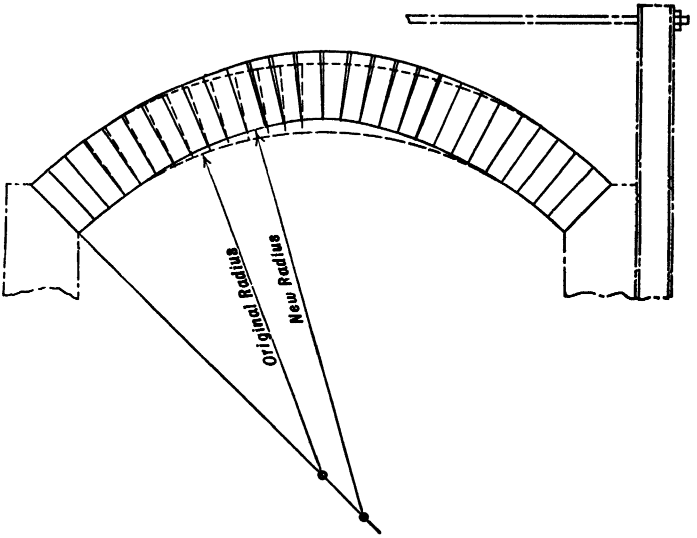

Fig. 207. Rising of an arch caused by heat in furnace.

Before the interaction of arch and binding can be taken up, a study must be made of the effect of temperature changes on arches. Between room temperature and furnace temperature of 2200 F, fireclay bricks expand about 0.075 in. per ft, or 0.6 per cent. The inside of the arch becomes longer by that amount, while the outside maintains practically its original length. Several things can happen under the influence of this expansion. The abutments can be pushed apart, the bricks can be compressed, or the arch can rise. It will be shown below that, in most heating furnaces, the abutments remain fixed or practically so.

Let it be assumed first that the bricks are incompressible. Then the arch rises, as shown in Fig. 207. Near the abutments, the bricks are in contact over their full length, but near the crown of the arch they tend to gap at the outside and touch only at the inner edge, which action shifts the line of force to the inner edge of that section. (This often results in the form of mechanical spalling called "pinch spalling.") In this condition, that part of the arch would be unstable, and could be swayed and moved like a hanging chain. However, as the disturbance stops, the chain regains its original position, but the middle section of the arch keeps on going further away from equilibrium position—either until it collapses or until some of the bricks touch further out toward the top surface, thereby changing the line of force to a more stable position.

In reality, conditions are not quite as bad as described, because the softening of the bricks at furnace temperature permits elastic in addition to plastic compression of the bricks on the furnace side. This action distributes the load over a certain surface until the unit pressure has become so low that the bricks no longer deform. The line of force will, for that reason, not follow the inner edge of the arch. It will lie very close to the edge near the center of the span, but will move toward or beyond the center of the bricks as the skewbacks are approached. The latter are rigid and limit the movability of the bricks near them, which must touch along their whole length. The line of forces will, therefore, be flatter than the arch itself (see Fig. 208).

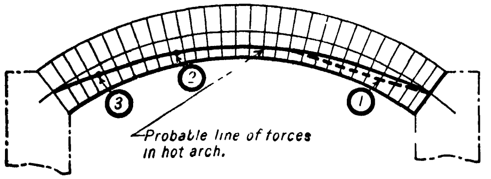

Fig. 208. Line of thrust in a hot arch.

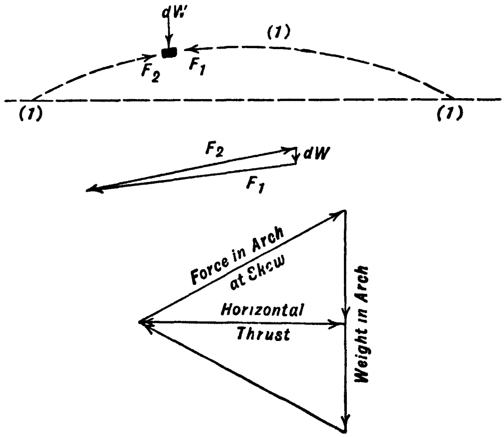

Fig. 209. Force polygon of an arch.

Before these facts can be used for practical calculations, the distribution of forces for a given line of action should be reviewed. In Fig. 209, let 1–1–1 be the line of forces in a given arch. The weight dW

of an element of the arch must be in equilibrium with the arch-thrust forces F_1 and F_2. In the center part of the illustration, the pointed triangle of forces, dW, F_1, and F_2, shows how the forces on each element are connected. Summations of the little triangles furnish the larger triangle, from which the horizontal thrust and the forces at the skews can be read for a given weight of arch.

But, as before stated, the location of the center line of forces within the arch is quite uncertain and varies, not only with the temperature, but also with the flatness or closeness of fit of the brick surfaces, and the elastic and plastic properties, as well as with the coefficient of thermal expansion. For that reason it is desirable to make an assumption that is simple and yet comes sufficiently close to the truth to serve for all practical purposes. It may, for instance, be assumed that the line of the arch forces follows the center line 2–2–2 (see Fig. 206) and that the actual forces are greater than those calculated on this assumption, a multiplication factor of 2 being used for furnace temperatures up to 1600 F, a factor of 2½ between 1600 and 2000 F, and a factor of 3 between 2000 and 2400 F. Or else it may be assumed that the center line of forces follows the line 2–4–2 in Fig. 206. This line joins the center line of the bricks at the skews with the inside edge of the brick at the center of the arch.

With the first assumption, the unit stress in the arch depends on its span, its rise, and the density of the arch material; with the second assumption, the unit stress in the arch grows with increasing thickness, provided that in neither case the arch carries any superimposed load. Although the second assumption takes cognizance of the flattened shape of the center line of forces, it leads to impossible stresses in very thick arches and is, therefore, discarded as useless. For calculation purposes, the other assumption is recommended,* although it likewise leads to impossible conclusions, for every furnace man knows that a long-span arch of only ⅛-in. thickness (for instance) could not be made to stand up. This failure is partly due to the movement of the abutments, as explained below, but is mainly caused by the instability mentioned on page 293.

It should again be noted that the assumption of the force lines coinciding with the center line of the arch and the recommendation of the force magnification factor were introduced solely for the purpose of calculation of the thrust of the arch. In reality, the line of thrust follows a different curve, as indicated in Fig. 208. In flat arches, and with certain kinds of bedding of the arch bricks, the thrust line may assume the shape that is dotted in the right-hand side of that illustration. In that case, the arch separates at point 1 and the center part of the arch slips into the furnace. Of course, the bricks near the skews fall in immediately afterward, except in circular vaults, or turtleback arches. In these the ring shape prevents the rest of the bricks from falling in, and for that reason, this peculiar accident can be studied in these arches. The flattening between points 2 and 3 (see left-hand side of Fig. 208), which usually precedes the reversing of the curve, as at 1, is a warning signal.

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* This recommendation has been criticized, but no better (or more practical) method of calculation has been proposed up to the time of this edition. Compare "Sprung Arch Roofs for High Temperature Furnaces," by J. S. McDowell, *Blast Furnace and Steel Plant,* June and September 1939, and February 1940.

If an arch has been partly burned or spalled away unevenly so that it is thin on one side while the other side retains its original thickness, its stability is endangered and collapse is quite probable.

It may be stated, however, that roofs of heating furnaces seldom fail from crushing of the arch bricks. Occasionally a very hot luminous flame melts an arch down; in other cases, ash is blown against the roof, sticking to it and acting as a flux which causes the roof to melt; but, as a rule, the rather rare collapsing of an arch is due to other causes, the most common of which is the yielding of the abutments (skewbacks), and pinch spalling (page 293). The skewbacks yield for several reasons. Poor foundations, giving way in spots, withdraw support from the arch at these points. Sidewalls, on which the arch rests, occasionally fail. In still other cases, the tierods or buckstays become overheated, yield, and allow the arch to change its shape. Occasionally, the furnace binding yields, because it is too weak even without being overheated. Destruction of furnace roofs is frequently due to spalling (see pages 283–284). After each cooling and rapid heating, the lower ends of many of the arch bricks are found to have cracked off and fallen out; the projecting pieces next crack off; and this piecemeal destruction proceeds until the stability of the roof is endangered.

As before mentioned, and as shown in Fig. 203, firebrick expands about 0.075 in. per ft. An arch, on being heated, must either rise or be compressed at the hot face, or stretch the binding. If the latter occurs with plastic deformation, the arch is doomed because, at each cooling, the arch sinks down to a flatter shape and, at the following heating, it stretches the binding a little more. Movements of this sort cause the arch to collapse after a limited number of heating and cooling periods. The abutments must be fixed and the arch must be allowed to rise, or else the binding must yield elastically. It will be interesting to note that the use of springs on tierods was patented by

R. Wells in 1856, with the idea of permitting the binding to yield from the excessive forces of expansion and return to its original position on contraction during cooling. The use of springs was common during the early years of this century, but furnace builders have now eliminated tierods and springs, and tie the buckstays rigidly together by means of structural channels (see Fig. 205), usually at the bottom as well as at the top of the furnace.

In connection with the moving of abutments, it should be noted that the smallest safe thickness of an arch is a function of the span and of the rise. It is quite common to make the radius of curvature of the inside of the arch equal to the span, which proposition makes the rise equal to 13.4 per cent of the span. The rise may be more but it should not be less than this figure, unless there exist some compelling reasons. For high temperatures, manufacturers of refractories recommend making the rise 2½ in. per ft of span, or 21 per cent of the span. The flatter the arch, the thicker it must be for a given span, to counteract possible dislocation of the skews. With regard to the relation between span and thickness, as dictated by the same possibility of movement of the abutments, nobody can tell just what the movement is going to be. All we can say is that there is more likelihood of large displacements in large furnaces than there is in small furnaces. In consequence of this uncertainty, practice varies and the following figures can offer a general guide only for firebricks. When using lightweight fireclay bricks, somewhat thinner arches are frequently used than are indicated by these figures:

Span less than	4 ft	12 ft	18 ft	24 ft
Brick thickness	4½ in.	9 in.	13½ in.	18 in.

In some cases it is desirable to make the bottom of the arch flat, instead of curved. This can be done, as shown in Fig. 210, by using somewhat longer bricks in the middle of the span. The construction is known as a "jack arch." It is evident that the stability of the arch and the force actions in the part above the lower curved line in the illustration are no different from those shown in Fig. 208 and that the only effect of lengthening the bricks near the middle is to add somewhat more load at that place. Some jack arches are flat not only at the bottom but also at the top. All bricks are then made of sufficient length to insure stability. Such an arch cannot rise so easily as a sprung arch upon being heated (compare Fig. 253). Occasionally, it becomes necessary to effect an abrupt change in the height of the roof at a particular place in the length of the furnace or of a passage. In a sprung arch, this can be done by constructing the arch as in

Fig. 211. The lower edge of each ring should, in that case, always be well below the level of the upper edge of the adjacent lower ring, in order that the longitudinal forces may be transmitted without tilting the bricks in a vertical plane; furthermore, it is well to have the projecting lower edges chipped off, as shown in the dotted line, to present an even inclined surface to the action of the flame.

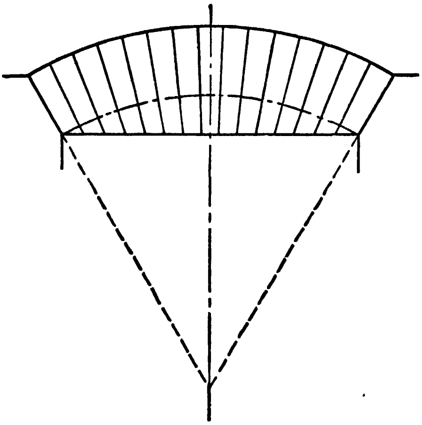


Fig. 210. Jack arch.

Among the causes for the collapsing of roofs are mechanical injuries, such as explosions or impacts. It was previously mentioned that furnaces are occasionally filled with explosive mixtures of air and fuel and are then lighted; the best of firebrick and the best design are of no avail against this treatment. Likewise, furnace roofs cannot be designed to withstand bumping by heavy ingots suspended from cranes, or other mechanical abuses.

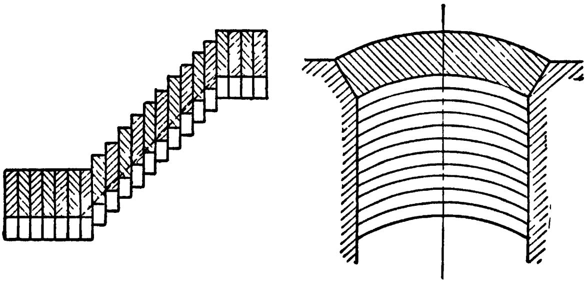


Fig. 211. Sloping arched roof.

In connection herewith, it should be explained that arches are built up of rows in which several straight bricks are alternated with one or more arch bricks (see Fig. 212). Brickmakers cannot, and do not, make arch bricks to suit every conceivable radius. In consequence, we must take the nearest brick with radius smaller than the arch, and must fill in with straights, the relative number of which is readily obtainable from tables given in most brickmakers' catalogues. A bad feature of this practice is that the straights are not held by wedge action, and could slip out if it were not for the frictional hold which the arch forces exert. Conditions are even worse when the bricklayers put wedge bricks upside down. In spite of obvious troubles which will be caused thereby, this is occasionally done.

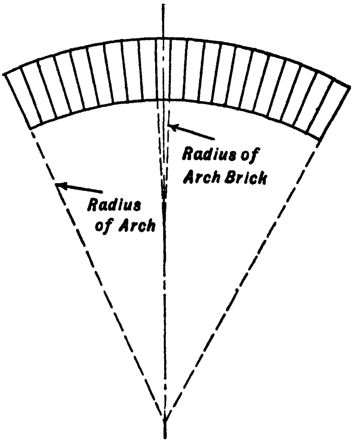


Fig. 212. Arch built up of arch bricks and straights.

The use of bricks, the faces of which are convex instead of flat, is to be avoided, because the bricks can rock or roll on the opposing surfaces. In laying an arch, each brick should be tested for flatness and any convex ones rejected.

Movements of the arch and vibrations due to panting flames occasionally overcome the friction hold and endanger the stability of the arch. Introduction of cold air into hot furnaces and repeated heating and cooling are injurious to roof arches, partly because of spalling and partly because of creeping and slipping of the bricks.

All heating furnaces are operated under a slight pressure. This pressure is greatest directly under the arch, because of the buoyancy of the hot gases. In addition to the heat transmitted by conduction, excess heat is carried into the more or less porous brickwork by these gases, resulting in higher temperatures throughout the arch than would be obtained in a tight roof. For this reason, leaky roofs of furnaces are to be avoided. The advantages of using furnace cements instead of fireclay for grouting the roof are discussed on page 337, but it may be well to mention here that a poor cement containing fluxes may ruin a roof made of the best brick, whereas a really good cement will increase the life of the roof.

The strength of buckstays and binding necessary to take the thrusts of arches is dealt with on page 355.

Suspended Roofs. Since the year 1920, the use of suspended roofs has grown rapidly, especially for spans greater than 14 ft. Special shapes of bricks are used, and the tiles are suspended from steel girders to form (usually) a flat roof. Many different kinds of suspension are in use, and their number is still growing. Suspended flat roofs are often (although incorrectly) called suspended arches.

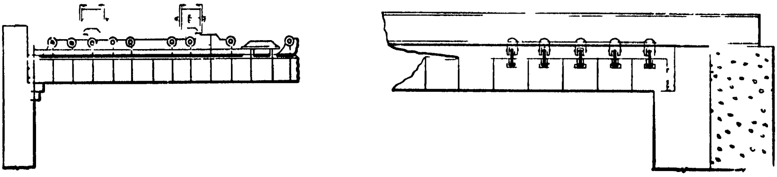


Fig. 213. Suspended roof (Detrick type)

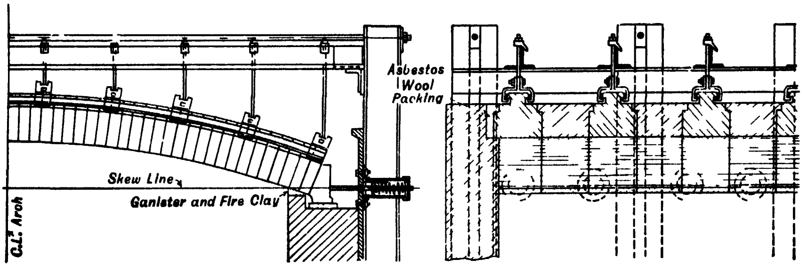


Fig. 214. Suspended arched roof (American Arch type).

The general design consists of (usually metallic) hangers which are suspended from overhead beams and which, with their lower ends, fit into holes or slots in the roof tiles, or grip projections on the tiles. Such designs are shown in Figs. 213 to 222. Most of the illustrations are clear enough to require no comment. In order to reduce the number of individual hangers, the roof tiles are made as large as manufacturing conditions (freedom from internal stresses) permit. In some designs, crossbars carry the tiles with occasional suspension of the crossbars from overhead beams. Because of the large number of different designs, standardization of the shapes of roof tiles is out of the question. The furnace builder must select the roof design that promises to be suitable, inexpensive, and accessible; he may or may not receive tiles of sufficiently refractory material. It is to be hoped that practice will eliminate a number of types and that finally only a few of the best designs will survive.

Several features are peculiar to suspended roofs, and they must be discussed. One is the absence of inherent crosswise pressure. Generally speaking, this absence does not differ from the absence of horizontal longitudinal pressure at right angles to sprung arches. Most designers depend on setting the tiles tightly at the time of installing the roof, providing a slightly compressible packing material at the ends and letting the expansion of the tiles against an abutment furnish enough pressure for making the roof tight.

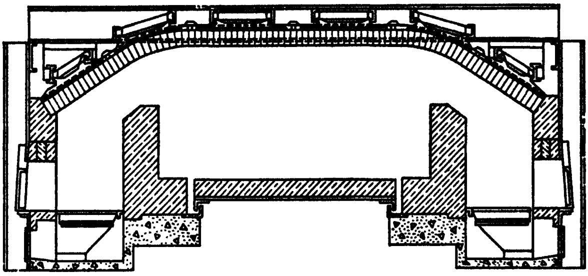


Fig. 215. Suspended roof over a sidefired car-type furnace.

There is a difference between slightly leaky sprung roofs and slightly leaky suspended roofs. Products of combustion escaping through a sprung roof can do little harm. Matters are different with a suspended roof. Sulfur dioxide in the products of combustion corrodes the steel in the hangers, especially if the steel is alloyed with nickel. If the roof tiles are covered with thick insulation, the gases become cool enough to condense the moisture in the products of combustion, and sulfuric acid is formed. It attacks the hangers very rapidly and causes sections of the roof to fall in.

The next item is difficulty with insulation. Most of the illustrations in this book show bare tiles. Thick insulation raises the temperature of the upper surface of the tiles to almost furnace temperature. If thick enough, insulation causes overheating of the hangers, which are then stretched by the weight of the tiles. Hangers of alloy steel permit the application of thick insulation. Refractory hangers have been introduced for the purpose of permitting the use of thick insulation. Figure 216 illustrates roof elements with a refractory hanger. Half-moon lugs on the hanger fit into depressions of the roof tiles. Usually, two hanger-supported tiles carry a wide tile between them. Figure 216*a* is a view of a roof assembly. Average firebrick creeps at 2200 F and with 25 psi about ¾ per cent in 100 hours. In a work year of 6000 hours it would amount to more than 14 per cent. The conclusion is that heavily insulated roof tiles must be made of superduty quality.

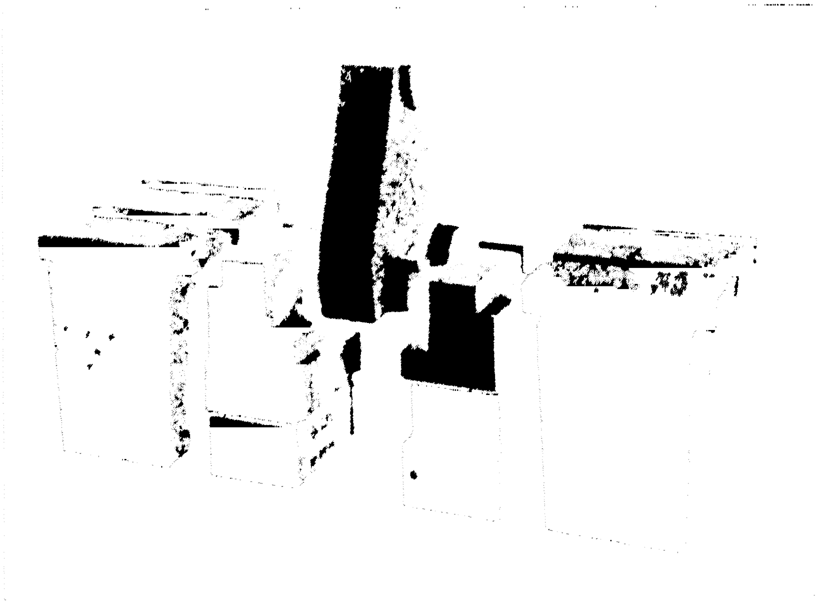


Fig. 216. Basic tile components of suspended roof with refractory hanger.

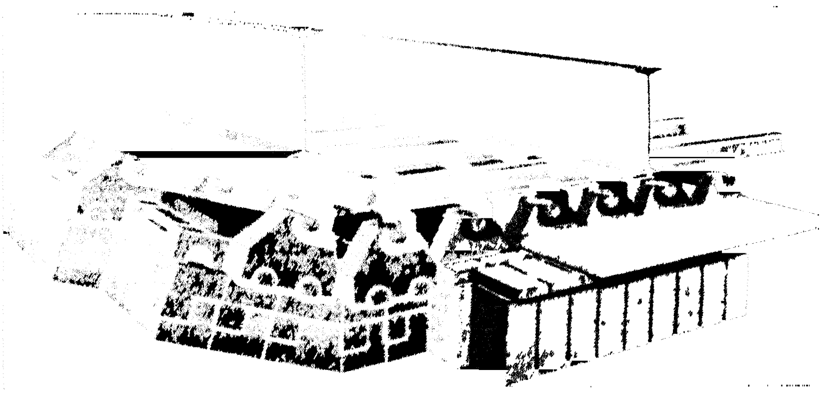


Fig. 216a. Assembly of tile-suspended roof. (Courtesy of Laclede-Christy Company.)

Another difficulty arises when repairs must be made. With several designs, a large number of tiles must be removed in order to replace a cracked tile. The tile arrangement illustrated by Fig. 217 permits a quick and easy repairing, but the water-cooled support pipes abstract much heat from the furnace; in addition, they encourage spalling.

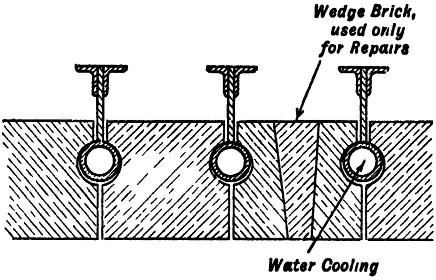


Fig. 217. Roof tiles suspended from water-cooled beams.

An important feature is the comparative ease with which a suspended roof can be connected to an overlying vertical wall. The design here referred to is shown in Fig. 218. Suspended noses are quite old in boiler furnaces. They were first adapted to industrial furnaces in connection with multiple zone heating. The details of the nose are

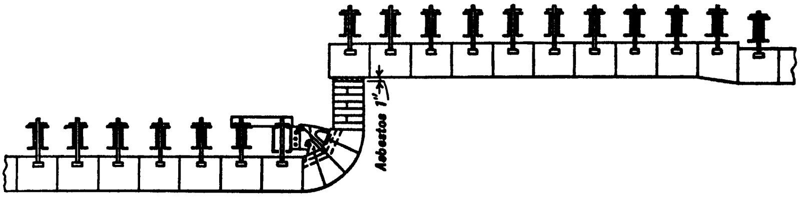


Fig. 218. Flat suspended roof, showing nose construction.

shown plainly in Fig. 219. A study of the illustration reveals the fact that a considerable number of tiles must be removed if one of them spalls severely enough to require replacement. All makers of suspended roofs furnish nose designs which are very similar to that of Fig. 219.

A few remarks on special designs are in order. Figure 220 illustrates a design by means of which the height of the tiles can be adjusted. Although the range of adjustment is small, it is enough to compensate for sagging of the overhead pipe and for irregularities in the tiles. Figure 221 presents interesting features. Each tile has a triangular groove in one side and a corresponding triangular ridge on the opposite

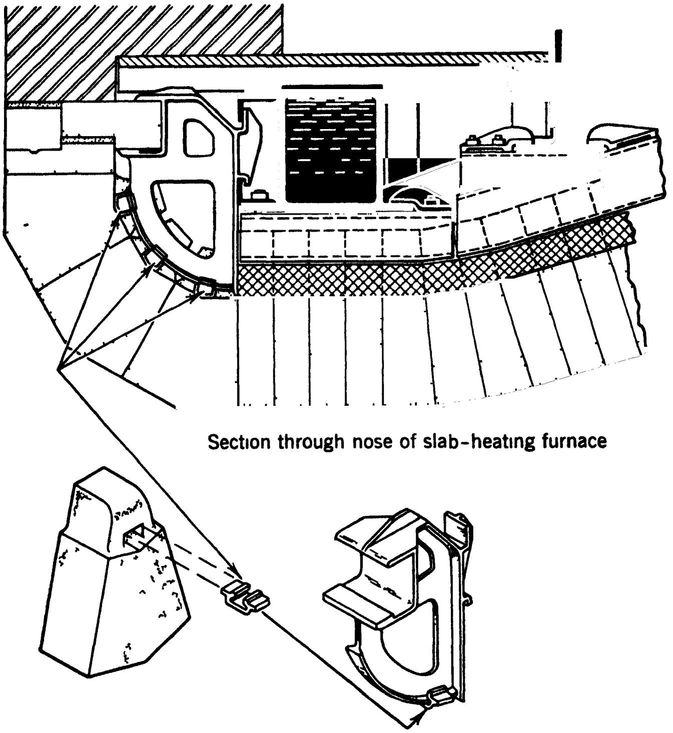


Fig. 219. Details of nose suspension

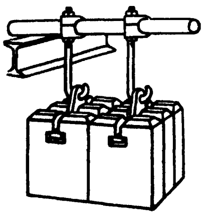


Fig. 220. Method of suspending roof tiles, permitting adjustment of height

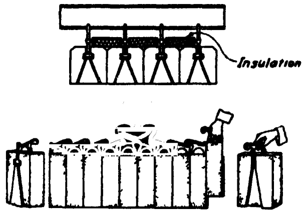


Fig. 221. Interlocking roof tiles, method of suspension and application of insulation

side. If the brick has been cracked by spalling, its bottom part cannot drop, because it is supported by the adjacent tile. The interlocking grooves and ridges also reduce gas leakage. The second feature is open space between insulation and hangers. While this space reduces the effectiveness of insulation, it prevents the condensation of corrosive vapors on the hangers. The tiles are supported in groups of 12. Those directly under an I-beam are hung from its lower flange. Each adjacent tile is supported from its neighbor by specially shaped castings of malleable iron.

An extremely simple method of support is illustrated in Fig. 222. This design was developed by the Babcock and Wilcox Company for the suspension of lightweight firebricks. The bricks (of standard size) are strung on a ⅝-in. rod like beads on a string. Every fifth to eighth brick has a lateral depression (as shown in mark 2) into which a bent suspension rod fits. Although excellent for lightweight bricks, the design cannot be recommended for dense firebricks. However, it appears that dense firebricks are "going out."

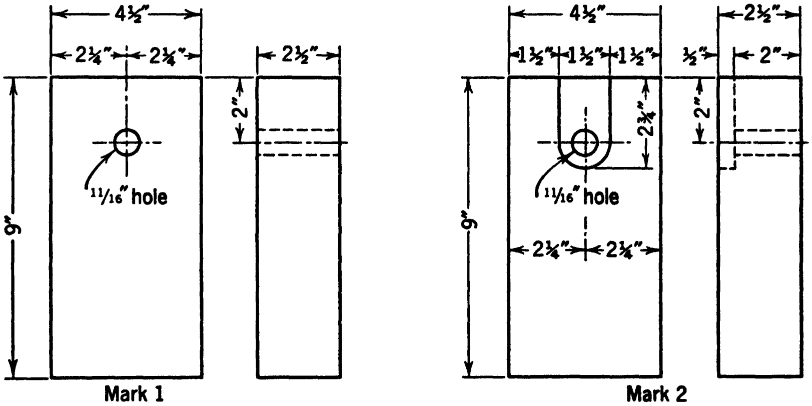


Fig. 222. Simple suspension method for lightweight roof bricks (Babcock and Wilcox type).

A type of roof which under certain conditions is self-suspended appears in Fig. 223 (Reintjes type). The roof is not quite flat, but has a slight rise; the skewbacks make contact only along the lower part of the tile thickness. Hence, under most conditions, the tiles are in compression; the action is that of a true arch, and the interlocking lugs on the tiles are not brought into action except if the line of force 1–3–2 should pass outside the middle third of the height of the tile, causing a tendency to open the joints. The lugs have sufficient clearance that the blocks can rock on each other to some extent. On cooling, as the bricks contract and the roof lowers slightly (dot-dash line in Fig. 223), the split key opens to a slight extent at the bottom, and the line of force shifts to 1–4–2. Only under extreme conditions, as when excessive shrinkage of the bricks has occurred or the skewbacks have moved apart, are the interlocking lugs called on to support the full weight of the arch; in such a case, each half of the arch may become a cantilever, each tile hanging from its neighbor nearer the wall, *provided* there is enough weight of bricks on top of the skewback tiles to hold these in place. The interlocking also reduces leakage of flame or gases through the joints.

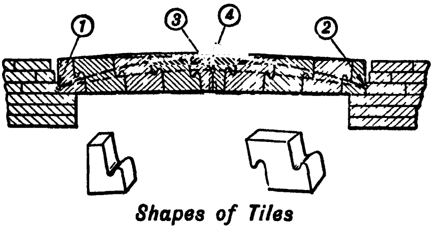


Fig. 223. Interlocking arch bricks (Reintjes type).

3. THE STRENGTH OF HEARTHS

Turning from the upper to the lower boundary of a furnace, we meet the subject of hearths and foundations. The strength and durability of these parts are of considerable interest to furnace users. It will be advisable to study first a hearth that also serves as a roof for a furnace cavity below the hearth. Such an element is used in underfired furnaces. In this type (see Figs. 224 and 225), the intention is to heat the material lying on the hearth, Fig. 224, very uniformly by transmitting heat to it from all directions. Heat travels up through the hearth in direction 1; it is radiated to the stock as indicated by arrows 2; and is conveyed to it as indicated by arrows 3. In this furnace, the hearth serves two purposes. It transmits heat and it carries load. The two conflict, because the hearth should be thin for transmitting heat and thick for carrying load. An idea of the extent

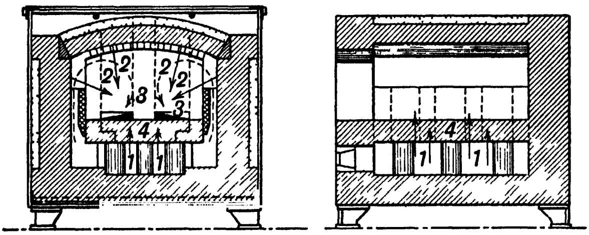
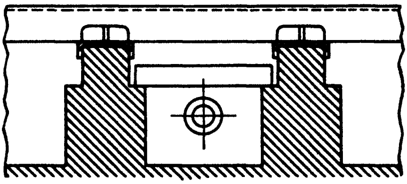


Fig. 224. Flow of heat in an underfired furnace.

of this conflict can be gained from the following analysis. Average furnace practice for the heat-treating of steel consists in heating an average of 30 lb of steel per (sq ft of hearth, hr) in an underfired furnace. If we assume that the steel is to be heated to the rather low temperature of 1400 F, the heat to the steel on each square foot of hearth will be 30 times 220 Btu per hr (see Fig. 28), or 6600 Btu per hr. The maximum permissible temperature under the hearth in such furnaces (with silicon carbide supporting piers) is 2100 F, so that the temperature difference in this case is 700 F. The hearth may be constructed with flat tiles spanning between piers, or with sprung arches between the piers. In each design, the maximum span between piers is about 12 in., and the minimum thickness, if constructed of firebrick, is 4½ in.



Partial section showing piers

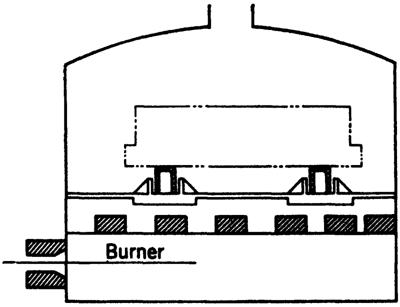


Fig. 225. Underfired furnace with perforated hearth.

On the basis of a heat conductivity of 10 Btu in. per (sq ft, hr, F) which is rather high for firebrick, the maximum heat transmitted through the hearth, if 4½ in. thick, will be $\frac{700 \times 10}{4.5}$, or 1550 Btu per (sq ft, hr). From this analysis, it is evident that the usual furnace hearth, with sufficient strength to support a reasonable load, transmits only a small fraction of the total heat required by the stock. If spans greater than 12 to 15 in. are required, the arches must be so thick that the advantages of a heat-transmitting hearth are greatly reduced.

These facts are well known to most furnace builders, and various means have been employed to overcome the limitation. Perforations of the hearth, as indicated in Fig. 225, reduce the temperature difference, because the openings allow a large part of the heat in the combustion chamber to radiate freely into the heating chamber, which lowers the temperature under the hearth by increasing the flow of heat from this area. Although the present chapter deals principally with the strength of furnaces, the following statement is in order: Material that is heated on a perforated hearth should not rest directly on the hearth, but should be elevated on rails or other devices. If this expedient is followed, very uniform heating can be obtained. If, on the other hand, the stock is to rest directly on the hearth of an underfired furnace, it is necessary to make the hearth solid, but to use a refractory of greater thermal conductivity. Among the various refractories that may be used for this purpose, silicon carbide stands out pre-eminently. Its thermal conductivity is 7 to 10 times that of firebrick, and its mechanical strength at furnace temperatures is about 12 times as great.

At first thought it might appear that the hearths of all other furnaces (in contradistinction to those of the underfired type) would always be strong enough, because they rest on a firm foundation. Nevertheless, their strength presents many problems. Comparatively little trouble is experienced with hearths of furnaces that carry a temperature of 1400 to 1600 F, except, as above stated, for underfired furnaces; but difficulties arise in furnaces with temperatures in excess of 2200 F, because of the chemical action of hot iron oxide on hot firebrick. For hearths of annealing furnaces and of heat treating furnaces, firebrick, particularly if dense or vitrified, is a good material, and no trouble arises with the hearth unless the latter be too thin, as will be explained later. In furnaces which heat the charge to 2000 F or higher, firebricks are frequently used as material for hearths. Whether or not firebrick is serviceable for hearths in such furnaces depends on the rate of heating. If the rate of heating is low (furnace temperature equal to 2300 F), the scale does not melt and does not form a slag with the silica in the firebricks. But, if the furnace temperature goes up to 2400 F or even higher, the outer skin of the scale melts (or softens), combines with silica, and forms a slag. If the hearth is ventilated, the cooling action from below chills the slag, after a sufficient amount has been accumulated. In that case, the hearth lasts for some time. If the hearth is not ventilated, it becomes a mushy mess.

In very hot furnaces, the slag becomes so thin that it runs off, if the hearth has a slight inclination. This circumstance is utilized for keeping the hearth free of slag, by inclining the hearth, and by draining the hearth through a slag hole especially provided for that purpose. The slag hole must be kept open either mechanically, by poking, or thermally, by the external application of heat. The trouble that is experienced with firebrick hearths may be somewhat reduced by making the bottom of silica sand. This also forms a slag with the iron oxide, but it can be more easily removed and replaced by a new bottom. An additional reason for the use of the sand bottom is the cheapness of the material.

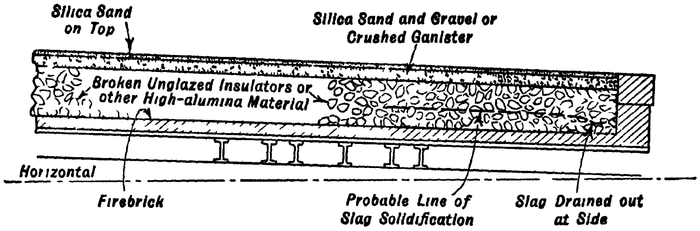

Silica Sand on Top
Silica Sand and Gravel or Crushed Ganister
Broken Unglazed Insulators or other High-alumina Material
Horizontal
Firebrick
Probable Line of Slag Solidification
Slag Drained out at Side

Fig. 226. Hearth of skelp-heating furnace.

In skelp-heating furnaces for butt welding and in lap-welding furnaces, the hearth is constructed as shown in Fig. 226. In these furnaces, the stock which is to be welded is heated until it is dripping The drops of molten steel or iron, and scale, falling on the hearth, would solidify and build up, if it were not for the presence of the sand and gravel, which are practically pure silica, and which form with the scale an iron silicate of lower melting point, that can run off easily. Below the sand and gravel, broken insulators or other lumps of high-alumina material, which resists attack by the slag, act as a French drain, allowing the liquid slag to flow toward the lower end where it is drained off. There must be sufficient thickness of material underneath to maintain temperature, so that the slag does not solidify, at least in the upper part. The sand and gravel are renewed about every five hours, and the aluminous material at much longer intervals. This type of hearth is not required in continuous skelp furnaces (Fig 74), because the skelp does not rest on the hearth.

As previously stated, chrome ore is frequently used on top of firebricks. It is neutral and is not attacked by scale, nor does it attack the firebrick (see also page 287). Figure 227 shows the grooved hearth of a continuous furnace, over which the billets laid on edge are pushed; the material of the hearth is chrome plastic, rammed in molds to form the grooves. In modern slab-heating furnaces, operating at high temperatures, high-alumina bricks are commonly used for the hearth of the soaking zone.

Hearth plates have already been discussed on page 262.

The above-mentioned softening of the whole hearth in high-temperature furnaces has led us to the use of ventilated hearths (see Fig. 228). The furnace rests on plates and girders or on piers between which the air can circulate. Even a moderate circulation of air will keep the bottom courses of brick fairly hard, and will prevent deep-reaching injury to the hearth. Calculation of cooling action is possible but rather tedious. Tentatively, this heat loss, for average ventilated hearths, can be taken as 60 per cent of what it would be if the surface were free and exposed. In the design of furnaces with ventilated hearths and in their operation, care must be taken not to block up the ventilation openings, because such blocking up would result in overheating of the supporting steelwork and in the collapse of the hearth. The ordinary rules of simple structural design are used in the calculation of the thickness, *s*, of the supporting plate; of the span, *L*, of the beams; and of the size of the beams. These rules need not be discussed here.

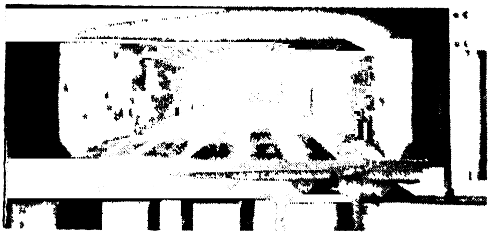


Fig. 227. Grooved hearth made of chrome plastic molded in place (Courtesy of Electric Furnace Company, Salem, Ohio.)

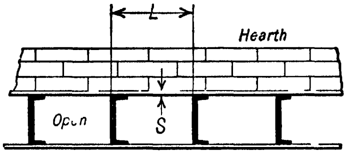


Fig. 228. Ventilated hearth.

On ventilated hearths, iron oxide cinder is occasionally used as a material for the bottom. If the cooling action is energetic, the lower surface of the cinder bottom, resting on the firebrick, will cool enough to save the brick from formation of slag.

Soaking pits are equipped with a very special hearth. The firebrick bottom is covered with a deep (about 18 in.) layer of coke breeze, which absorbs molten iron and slag. When the bottom has become saturated with slag, the mixture of coke and slag is pushed by the overhead crane to an opening in the hearth and drops into a slag car. If the fresh coke breeze is damp, the bricks spall at the hot slag line. Chrome magnesite resists the spalling action.

The hearths of medium-sized and small furnaces, say less than 40 sq ft of hearth area, are sometimes made solid and rest on concrete foundations. Troubles arise in this case if the firebrick or chromite hearth is too thin, or if tie rods ran too close to the top of the hearth. The nature of these troubles will become clear from the following discussion.

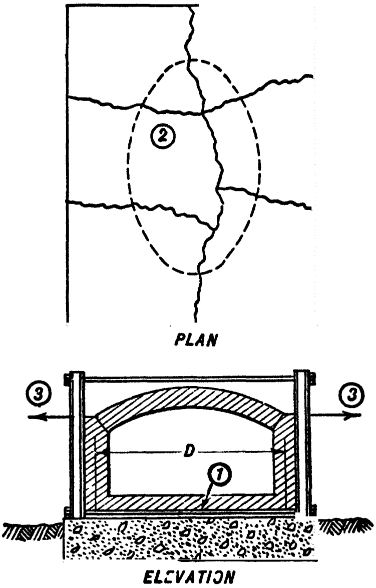


Fig. 229. Effects of thin hearth on concrete foundation.

Let Fig. 229 represent a forge or mill-type furnace (2200 to 2300 F) with a comparatively thin brick hearth resting on a concrete foundation. The bottom tie rods, 1, become overheated; they become so soft and weak that the arch thrust can stretch them. The roof collapses as the result of improper hearth design. The heat which is transmitted down through the hearth flows as indicated in Fig. 81, with the result that the center part of the concrete foundation, as indicated by 2, within the dotted outline of the plan of Fig. 229, becomes much hotter than the outer part and tends to expand. That tendency is resisted by the outer, cooler edge of the foundation. If the hearth be thick, temperature equalizes itself within the firebrick and no harm is done to the foundation. But with the thin hearth, the center of the foundation becomes so hot, compared to the edges, that the tensile strength of the latter is overcome, and the foundation cracks into several individual pieces similar to those indicated in Fig. 229. The position of the cracks naturally varies with the accidental location of flaws and planes of weakness in the concrete.

The load on the different parts of the foundation is never uniform, and the very purpose of a foundation is to take up the lack of uniformity, causing the load distribution on the subsoil to be practically uniform. As soon as the above-mentioned cracks in the foundation have traveled far enough inward, the different sections settle in different degrees, cracks appear all over the furnace, and collapse will occur in a short time.

In addition, thin firebrick hearths cause overheating of the concrete in the center. Concrete can stand high temperatures for a short time, during which only its surface is overheated, but it disintegrates if continually exposed to high temperatures. Dehydration of cement begins at 500 F and is completed at 900 F (special cements are now available that withstand somewhat higher temperatures). For that reason, a cement foundation is an unnecessary expense if, in operation, it is to be heated to 600 or 700 F, because it will, in that event, finally become loose fill, without any cohesion whatsoever. In that case, the only advantage of a concrete slab is that it provides a level surface on which to erect the furnace.

The question immediately arises: How thick should a non-ventilated hearth be to protect the foundation? Referring to Fig. 89, it is evident that the temperature of the foundation is highest at point 2. This temperature should not exceed 900 F. In Fig. 89, the dotted lines perpendicular to the hearth surface, such as 1–2 or 3–4, represent flow lines of heat; the solid lines, cutting each flow line at right angles, are lines of equal temperature.* The highest allowable temperature of point 2 is approximately

$$T_2 = T_o + (T_i - T_o) \,.\, e^{-1.92\frac{z}{D}} = 900$$

where z = thickness of hearth, D = width of hearth (inside the walls), T_i is the time average of the inside temperature, and e is the base of Naperian logarithms. From these conditions the following rather approximate rule has been formed, for intermittently operating furnaces: For a hearth of fireclay brick or sand of width D, as shown in Fig. 229, a thickness of $D/6$ is sufficient, if the furnace temperature does not exceed 2200 F; and $D/8$ is sufficient if the furnace temperature does not exceed 1600 F. In either case, D is the shortest dimension of the hearth. For continuous operation, unless the hearth is ventilated it must be considerably thicker than these figures indicate.

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* J. D. Keller, *Transactions American Society of Mechanical Engineers,* Volume 50, 1928, page 111.

Hearths over which material must be pushed have received much attention from furnace builders and furnace users. The pushing of heavy material across a sand or brick hearth is always injurious to the hearth, because unequal expansion, or spalling, or slag working between and under the bricks, produces sufficient unevenness to cause some part of the heating stock to catch, usually ripping up the hearth and leaving it in poor condition. For that reason, hearth bars, plates, or skids of some sort are usually provided when pushing heavy material more than 4 in. thick. In places where the only movement of the material occurs when it is laid down or pulled out of the furnace, brick piers or alloy blooms are provided to improve bottom heating by supporting the stock above the hearth, which also protect the hearth from damage.

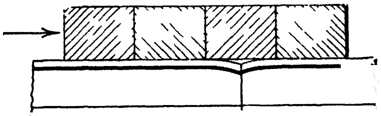


Fig. 230. Joint between hearth rails.

As previously mentioned, hearth bars and hearth plates of cast iron are used in furnaces where the temperature does not exceed 1600 F. On account of the growth of cast iron, these metal parts are preferably made in sections, with rounded corners at the joints, as shown in Fig. 230. Some designers are very careful to prevent rising or buckling of such metal parts by anchoring one end of each section and arranging a projection to hold down the adjacent end of the next section, with ample provision for expansion and growth between the sections, as shown in Fig. 231.

Difficulties, varying in degree, arise with hearths of continuous furnaces operated at temperatures from 1900 to 2300 F. In topfired furnaces heating small billets and thin slabs, "dry skids" are almost universally used to prevent billets from catching and tearing up bricks; tearups are caused by crooked billets and by "shear drag" at the ends of the billets. Such skids commonly consist of 4-in. by 6-in. slabs of steel or heat-resisting alloy in about 10-ft lengths, and arranged to lap each other at the ends to form a continuous support. The skids project 1½ in. above the refractory, and are often held at one end by a 1¼-in. bolt to the bottom plate of the furnace, as shown in Fig. 232. In such furnaces, the skids are usually carried only part way to the discharge end of the furnace, with the soaking portion of the hearth beyond the end of the skids constructed of plastic chrome mixtures tamped in place. Such hearths have given excellent results.

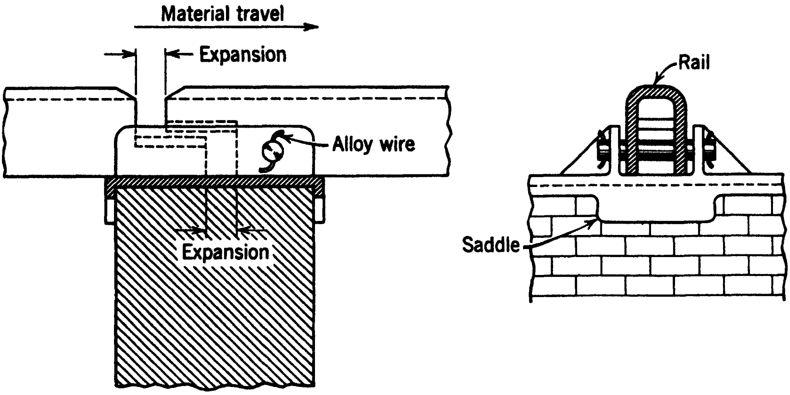
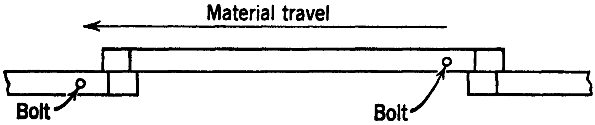
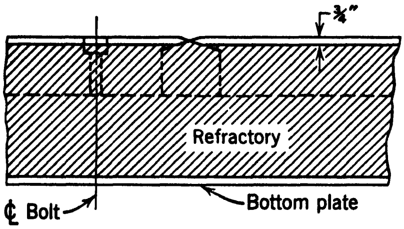


Fig. 231. Rail joint and anchor.

Another mixture consists of grain magnesite with ground open-hearth slag and a portion of the grinding dust formed in grinding steel billets, tamped in place wet. All hearths at the hot end of continuous furnaces require some attention at regular intervals in scraping out scale and preventing formation of lumps or mounds; if lumps should form, they



Plan view



Enlarged section

Fig. 232. Typical dry skids in refractory hearth.

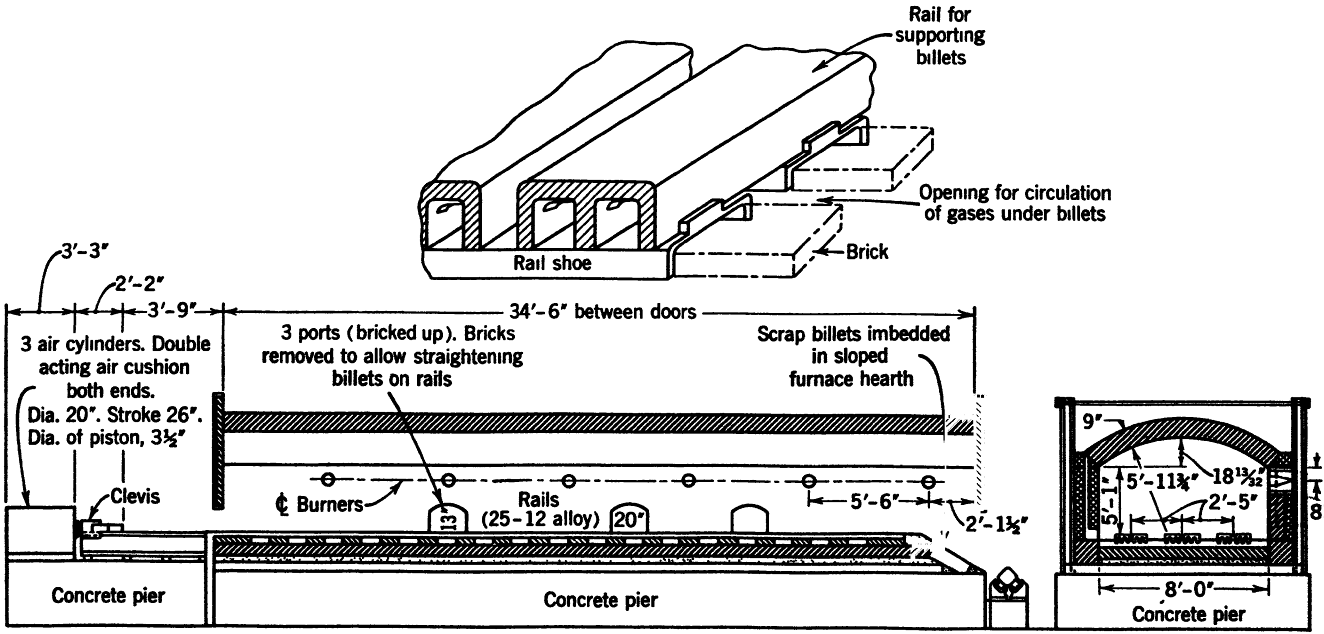

Rail for supporting billets
Opening for circulation of gases under billets
Brick
Rail shoe
3'-3"
2'-2"
3'-9"
34'-6" between doors
3 air cylinders. Double acting air cushion both ends. Dia. 20". Stroke 26". Dia. of piston, 3½"
3 ports (bricked up). Bricks removed to allow straightening billets on rails
Scrap billets imbedded in sloped furnace hearth
Clevis
℄ Burners
13"
Rails (25-12 alloy)
20"
5'-6"
2'-1½"
Concrete pier
Concrete pier
9"
5'-1"
5'-11¾"
18 13/32"
2'-5"
8½"
8'-0"
Concrete pier

Fig. 233. Furnace with high-alloy rails for heating heavy blooms.

must be sledged loose at the first opportunity in order to keep the hearth smooth.

If money is no object as, for instance, in the heating of billets for forging artillery shells, skid rails of high-alloy steel are sometimes carried all the way to the hot end of a continuous furnace. Figure 233 contains details and arrangement of a furnace and of hearth rails of alloy steel. The billets slide on wide rails which receive some heat from gases circulating under the rails. Furnaces of this design give no trouble while heating 9-in. by 9-in. blooms to 2250 F.

The soaking hearths of heating furnaces for heating thick slabs or large blooms are usually constructed of firebrick and insulation, with a top course of best quality high-alumina bricks. Here also, steel or alloy skids, or "dry skids," are used, consisting of 3- or 4-in. wide by 9- to 11-in. deep slabs, arranged as previously described. In practice, it has been found that all such dry skids have a tendency to rise, largely because of unequal expansion of the hot top surface and the relatively cold bottom surface. This can be prevented to a great extent by cutting the slabs part way through from the bottom surface, but the best plan is to keep the slabs covered by the heating stock in the furnace, even to the extent of staggering the charging of short slabs which are not of sufficient length to cover all skids in the furnace, so that weight is bearing on all skids in the furnace at all times.

Dry skids are sometimes made of a material which does not soften at 2300 F. Such material is clay-bonded silicon carbide. The usefulness of this material at the hot end of a high-temperature continuous furnace depends on the method of operation of the furnace. If, for the purpose of increasing the through-put, the furnace temperature is unduly increased or if flames impinge on the billets, the edges and corners of the slabs or billets melt. If molten steel comes in contact with hot silicon carbide, the latter is eaten away. The evil effects of this action are mitigated by laying the silicon carbide rails so that the top surfaces are flush with the surrounding chrome plastic. In the cooler zones of the furnace, the skids project above the hearth. Silicon carbide rails are 2 to 2½ in. wide and 4½ to 8½ in. deep. They are made in sections having a length of 9 to 18 in. and have interlocking tongues and grooves. They are illustrated by Fig. 234.

For the heating sections of large continuous furnaces heating heavy slabs and blooms, water-cooled skids (skid pipes) are standard. Such skids produce a cold spot on the heating metal, which is largely removed on the soaking hearth, where dry skids must be used, as previously described. Water-cooled skids have been in use for many years. Friction between the stock which is being heated and the skid pipes wears the latter, which for that reason, are made of double extra strong pipe, usually 2-in, or 2½-in. nominal size (see Fig. 235).

Another reason for the use of double extra strong pipe is the fact that total force exerted by the pusher must be taken up by tension in the skid pipes at the charging end.

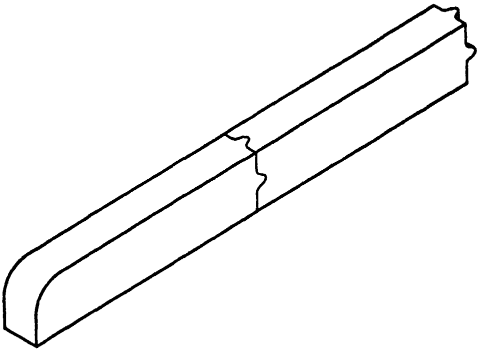


Fig. 234. Carbofrax hearth rail.

As the pipe wears thinner, heat losses to the water increase, and the danger of leakage increases. For that reason, some designers have screwed the pipes into a manifold, so that they can be turned, exposing new and unworn surfaces to the steel. Leakage at the screwed joint, sticking due to rust at the same place, and inconvenience of design with a manifold have caused this arrangement to be generally abandoned.

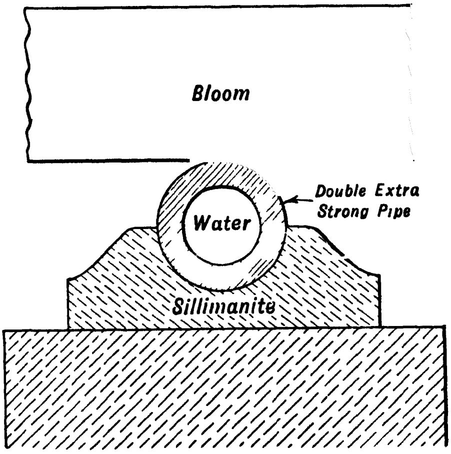


Fig. 235. Water-cooled skid pipe.

The ordinary pipe skid abstracts so much heat from the heating stock (by direct contact) that various devices have been introduced for the purpose of reducing this heat loss. Several of these devices have failed in practice, but others have survived. A common practice is that of welding wearing strips onto the top of the pipe (as shown in Fig. 236). The strip may be quite narrow; if a pressure of 125 psi be allowed between the stock and the skid (and that pressure is not excessive at 1200 F temperature of surface of skid), the width of the strip need only be equal to the thickness of the supported steel (inches) divided by 450. These strips are very hot at the top, tend to expand, and become loose unless the weld is very good. In the latter case, overheating of the strip may cause the pipes to buckle upward unless they are well anchored, and held down at all times by the weight of heated steel. A design that avoids this trouble is shown in Fig. 237; the welded strip is notched at short intervals, allowing the outer, hotter layers to expand freely. The notch must *not* be a saw cut; otherwise the strip and pipe will crack. As the friction and wear are greatest at the cold end of the furnace, the wearing strips sometimes extend only one-third or one-half the length from the cold end. This arrangement, however, produces excessive heat loss at the hot end and black spots on the steel. The life of these skids is 4 to 6 months, after which new strips may be welded on. After the second strip has worn off, the pipes are discarded, unless an earlier replacement was brought about by clogging or corrosion. The skids frequently fail by cracking where the pipe sections are welded end to end, as several pipe lengths are necessary to run through a long furnace.

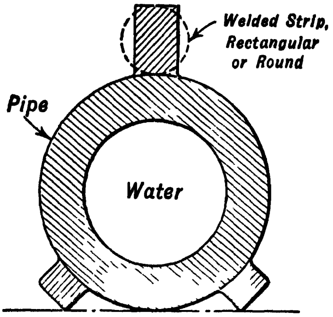


Fig. 236. Water-cooled skid pipe with welded strips.

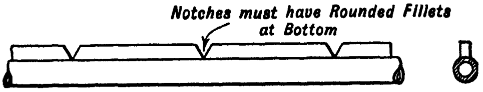


Fig. 237. Skid pipe with wearing strip notched to permit expansion.

The welded wearing strips are sometimes replaced by alloy welding rod applied to the top of the skid pipe and then ground smooth, particularly where scratching of the hot slabs must be avoided. By building up the alloy to a thickness of about ¼ in., satisfactory wear is obtained and the pressure is distributed in a somewhat better manner than with the usual wearing strips.

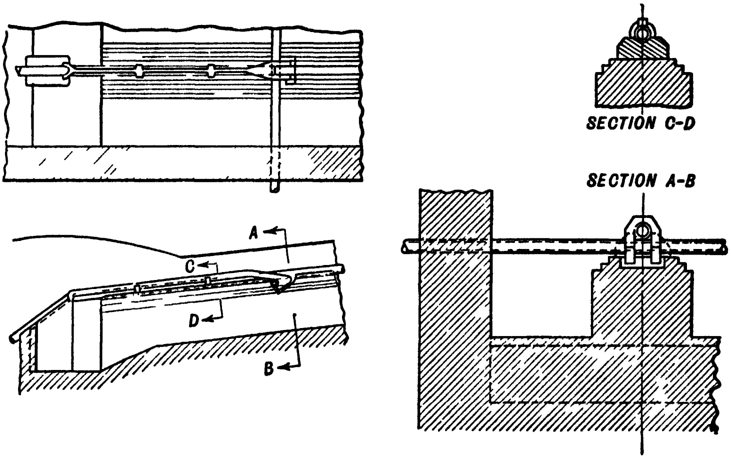


Fig. 238. Forked wearing strip, steel casting

One of the greatest troubles with the use of water-cooled skids is the failure of the water supply. Where possible, provision is made for turning water from the public water supply into the lines in case of failure of local supply. Very often the rules of the water company prohibit automatic switching over, for fear the public water supply be contaminated. In such a case, the installation of a very loud warning signal is advisable, which prompts the operator to shut off the local supply and to turn on the public supply before damage is done.

At the hot end of the furnace, a steel casting about 2 in. thick is sometimes used on top of the pipe. It wears or burns out in 3 to 6 weeks, and must then be replaced. A further development of this idea is shown in Fig. 238, in which the wearing strip is a steel casting which rests on top of the water pipe, is forked at the rear end, and hooks over a cross pipe which takes up the force due to the drag of the billets. By giving the skid pipes a reverse kink (down in section A-B, and then back up again parallel to the inclination of the hearth), the projection of the skid casting above the level of the skid pipes can be avoided.

Another arrangement consists of a number of thick, rolled-steel bars at the hot end, set on different centers from the pipe skids and slightly higher than the latter to eliminate black spots. These bars must be replaced frequently; they last 4 to 6 months.

In furnaces that carry a temperature below 1800 F, the wearing surface may be located at some distance from the water. Figure 239 shows a design that embodies this principle. A removable steel tongue is held in a water-cooled frame. The principle also appears in the use of a double pipe. A water-cooled pipe rests inside and at the bottom of a somewhat larger skid pipe. The latter is cooled by radiation and tangential conduction to the inner pipe.

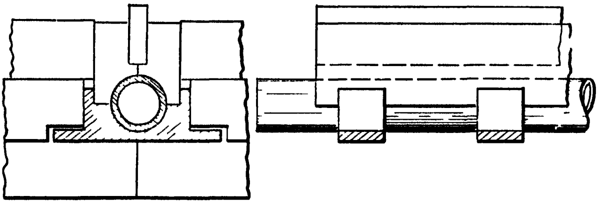


Fig. 239. Skid rail with depressed water pipe.

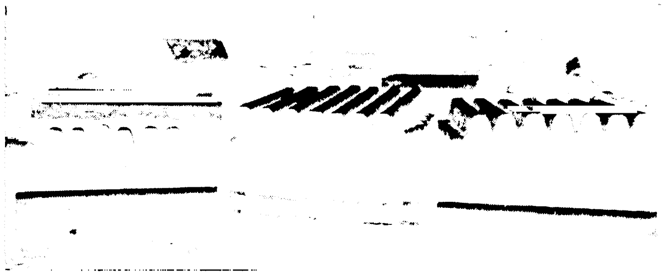


Fig. 240. Skid-rail castings of heat-resisting alloy.

Indirectly cooled castings of heat-resisting alloy used as skid rails are shown in Fig. 240. When in place in the furnace, they fit over the water pipes but do not rest on them. The separate pieces bear only at the ends. In spite of all care, it is extremely difficult to avoid warping. Tests for the purpose of doing away entirely with water cooling and of using alloy skids have been carried out here and there, but, up to the present time, only very wide alloy bars withstand abrasion near the hot end of furnaces that heat metal to 2200 F or above. Uncooled alloy skids serve well in the relatively cold two-thirds of the furnace length. Uncooled skids are avoided in high-temperature furnaces with top and bottom firing, because they receive high-temperature heat from the supporting rider walls and because water-cooled skid pipes must, in any event, be used near the hot end.

This discussion naturally leads to the strength of skid-pipe walls or piers on the hearths of continuous furnaces working with split flame. If these walls (see Fig. 241) are made tall enough to allow half of the products of combustion to pass under the stock, they can give trouble, particularly in large furnaces. There is practically no cooling action; outflow of heat can take place only in the direction of the arrows. The whole interior of the walls becomes heated uniformly and the bricks soften, unless built of the very best material. Even then it is a troublesome task to keep them in good condition, particularly if scale drops on the walls from the heated billets. This scale forms a slag with the firebrick and cuts the walls very quickly at furnace temperature. Attempts to cool skid walls by blowing air through a metallic box in the middle of the wall have proved unsuccessful, on account of slag working in between the box and the bricks and causing the latter to fall off. Another method, which has proved more successful, is to imbed a few water-cooled pipes in the wall. The skid walls last longer than without water cooling, and the heat loss is negligible.

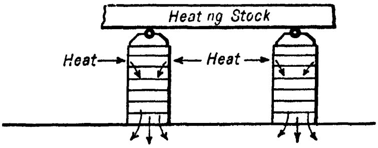


Fig. 241. Diagram illustrating the overheating of tall skid walls.

Non-cooled pier walls made of sillimanite or high-alumina bricks have been used successfully, resisting both softening and slag attack very well. For best service, large blocks of material are used, on which the skid pipe rests in smaller sillimanite carriers of special shape, or in alloy caps which are cooled by the water-cooled skid pipe. With high-alumina blocks, sufficient heat can be developed under the skids to heat large blooms and slabs uniformly from top and bottom.

The various methods of supporting the water-cooled skids in furnaces with top and bottom firing are illustrated in Fig. 242. There can be no question about the durability of any of these designs, as long as cool, clean water or boiler feedwater is circulated through the pipes. Heat losses to skids and vertical stanchions are greatly reduced, if they are surrounded by a refractory. Several different designs are available for this purpose, as shown in Fig. 243, sketches *a*, *b*, and *c*. The refractory rings in sketch *a* must be very thick to prevent spalling. A representative diameter is 12 in. In sketch *b*,

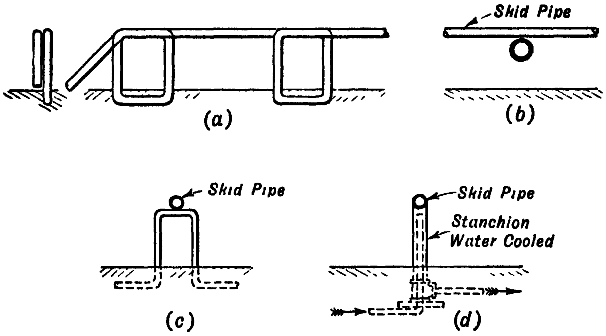


Fig. 242. Methods of supporting skid pipes in split-flame furnaces.

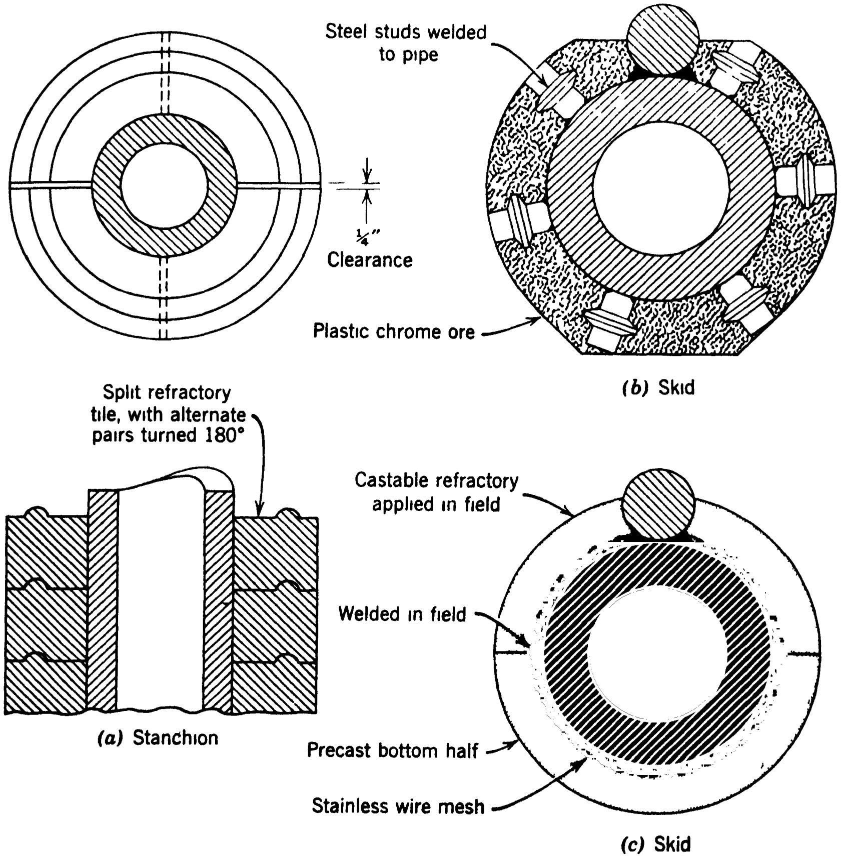

Steel studs welded to pipe
¼″
Clearance
Plastic chrome ore
(b) Skid
Split refractory tile, with alternate pairs turned 180°
(a) Stanchion
Castable refractory applied in field
Welded in field
Precast bottom half
Stainless wire mesh
(c) Skid

Fig. 243. Insulation of water-cooled stanchions and skids. Method (c) is also arranged with molded half-sections supported from welded studs at the top of the skid pipe.

plastic chrome ore is held in place by studs projecting from the pipe. In sketch *c*, stainless wire mesh is welded to the pipe to hold a castable refractory in place. These heat-saving devices are useful only if time and care in maintenance are devoted to them.

If the design permits, vertical stanchions are located directly under the skid pipes. If the design does not permit this arrangement, tubular crossbars, resting on the stanchions, support the skid pipes, as shown in Fig. 242*d*. The size of the crosspipes is computed from the well known rules on the bending strength of beams.

Insulation of horizontal pipes involves the same difficulties as that of vertical stanchions. The methods employed are the same as shown in Fig. 243, sketches *b* and *c*.

A great deal might be written about the strength and durability of the hearths of automatic furnaces, through which the stock is moved by chains, rocker bars, and other mechanical devices. However, designs of that character are more or less special, and no generally applicable rules can be given. The durability features of special hearths are discussed in the chapter on labor-saving appliances in Volume II.

4. FOUNDATIONS

The purpose of a furnace foundation, as stated in the preceding division, is to distribute non-uniform furnace weights uniformly to the subsoil. The loads that different soils can safely bear are given in handbooks and need not be repeated here. The total weight of "furnace, plus charge, plus foundation," divided by the specific bearing capacity of the soil, determines the area over which the foundation must be spread. The bending moments produced by the uniform upward thrust of the soil, and by the more or less concentrated downward loads of the furnace and charge, determine the thickness of the foundation. With furnaces resting on ventilated hearths, this procedure can be carried out according to the rules of mechanics; but, if the hearth of a furnace rests directly on the foundation, heat expansion, as explained previously under the heading of hearths, must be considered. The danger of cracking the edge of the foundation, by expansion of the center, has been explained, and it stands to reason that these dangers can be minimized by making the concrete foundation thinner in the center of the furnace than around the outside (see Fig. 244). The advantage of this difference in thickness is evident. The force produced by the heat expansion of the thinner concrete in the center will be small compared to the tensile resistance of a large rim of concrete. If heavy ingots, or similar concentrated weights are to be heated in the furnace, it is better to produce even pressure on the subsoil by a thicker hearth combined with a slightly thicker foundation than by a thicker foundation alone, because of the effects of heat expansion, as previously mentioned. It is also advisable to pack tamped and rammed cinder into the center part.

Solid hearths and solid foundations leave no cool place in which to locate bottom tierods. For that reason it is better to do away with them altogether, by burying the bottom ends of the buckstays in the foundation. Rather simply stated, the advantage of this arrangement is that the bottom tierods cannot become overheated, because there are none.

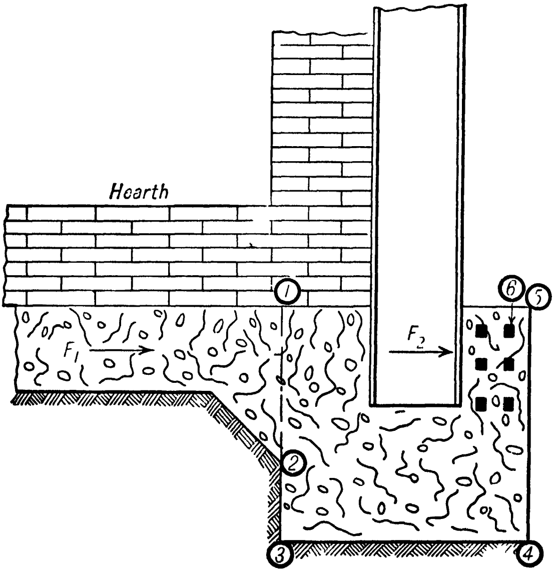


Fig. 244. Reinforced furnace foundation taking the place of bottom tie rods.

The device of burying the buckstays in the foundation must, of course, be used with care. If the foundation is cut up by flues, it cannot be depended on to transmit tension. In any event, it is safer to reinforce the outer ring with rods; see 6 in Fig. 244. The desirable amount of reinforcing steel can be judged from the fact that the outer ring of the foundation constitutes a beam of the cross section 1–2–3–4–5 and is loaded by the buckstay thrust, as well as by the expansion of the center. The shape of the furnace, the thickness of the hearth, the design of the roof, and the ratio of the thickness of the center of the foundation to that of the outer ring, all affect the amount of reinforcing required; but for average conditions, the following rough and ready rule gives good results: for every square foot of hearth, provide $\frac{1}{8}$ sq in. of reinforcing rod (in each beam or on each side of the furnace) for 2200-F furnaces, and $\frac{1}{12}$ sq in. for 1800-F furnaces.

It should be noted that the expansion of the hearth may crack a foundation of the type under discussion. It is advisable to lay the two upper courses of the hearth a little loosely, or to allow some room between the furnace walls and the binding at the level of the hearth, so that the wall may fit the binding snugly without excess force when the furnace is hot. If that precaution is not taken, the binding is bent out of shape, or else, if the buckstays are quite strong, the foundation cracks.

For large furnaces with areas of about 400 sq ft, the solid hearth must be made very thick, and too much reinforcing material is required in the foundation. The ventilated hearth is then preferred, and the foundation is kept free from temperature stresses. Small furnaces of not more than 20 sq ft length area are frequently built after the design shown in Fig. 244, but without the reinforcing rods, it being possible to put so much concrete into the ring that no danger of cracking the foundation exists. If the design of Fig. 244 is applied to a foundation through which a flue leads, it is usually necessary to restore the strength of the cut in the foundation by a plate or channel which spans the cut at the upper surface of the foundation and which is fastened to it by foundation bolts.

Foundations for rocker-bar furnaces and for other automatic furnaces are, as a rule, quite specially designed to secure accessibility of the mechanism under the furnace hearth. They are discussed in Volume II in connection with such mechanisms.

Continuously operated furnaces with large (not ventilated) hearths heat the sub-foundation soil to a depth of many feet. If the nature of the soil is such that it shrinks when heated, the foundation sinks in the middle, because the heat penetrates to the greatest depth in that region. If, before construction of the furnace, several feet of soil are removed and are replaced by slag, no injury is done to the furnace foundation.

Forge furnaces which are located near steam hammers are sometimes placed on springs for protection against vibration and shock. Figure 245 (taken from *Gas Trends,* January 1943) illustrates the principle. For complete protection, forge furnaces are enclosed in steel plates, and piping is arranged with flexible connections.

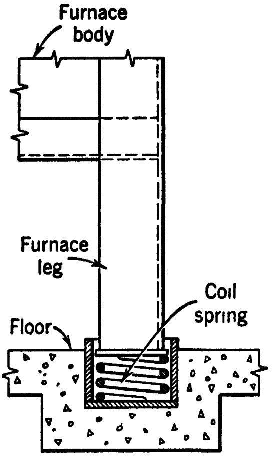


Fig. 245. Elastic support for drop-forge furnaces

5. SIDEWALLS, PARTITIONS, AND BRIDGE WALLS

Generally speaking, sidewalls of heating and annealing furnaces cause less trouble than roofs or hearths. Nevertheless, conditions arise here and there that lead to cracking, sagging, burning off, or collapsing. It will pay to study these troubles that occur most frequently.

Smooth walls, not interrupted by openings, stand up very well in small furnaces, even if the temperature reaches 2300 F. In large furnaces, various causes combine to produce annoyance. Very tall or very long walls of high-temperature furnaces are subject to a considerable difference in linear expansion between the furnace side and the outside. If the expansion of the inner layers can stretch the tie-rods, or binding, cracks open up in the outer layers. The cracks increase wall (heat) losses, but no structural harm is done by them, though the owner may imagine that the cracks are the beginning of the collapse of the furnace. If, on the other hand, the binding of the furnace is strong enough to offer considerable resistance to the expansion, several things can happen. If the bricks are of average quality, they are plastically deformed at the hot face. If the bricks are held rigidly, without expansion joints, and if the bricks are of the high-alumina type, they pinch spall at the hot face. If high-quality bricks are not held rigidly, and if no expansion joints are provided, the wall may buckle into the furnace.

On large furnaces, vertical expansion joints for horizontal expansion are often provided. The required width can be computed from the expansion data in Fig. 203. The joints are filled with cardboard or wooden strips when the furnace is being erected. Expansion joints are offset, as shown in the plan view (Fig. 246).

Buckling of a large wall in the furnace can be avoided by building the wall slightly concave on the hot face. In zonefired furnaces with underfiring, the hearth of the soaking zone tends to push the vertical wall in to the underfired chamber. Damage is prevented by making that wall arched.

Since the buckling tendency of walls of large extent is restricted by bonding, it stands to reason that the thickness of the bonded portion should grow with the extent of unsupported wall.* Since in any unsupported rectangle the shorter side determines the strength, this side, whether it be height or length, should be used for determining the minimum thickness of the firebrick wall, it being understood that protection against heat loss may require additional thickness. Tentatively, we may, for high-temperature furnaces, use 4½ in. of firebrick up to 3 ft of length or height, 9 in. up to 6 ft, and 13½ in. up to 10 ft. For 1600 to 1800 F furnaces, these dimensions may be exceeded 40 to 50 per cent. Greater unsupported lengths seldom occur; if they do, they require either 18 in. of firebrick, or else an auxiliary support for the wall, for instance, by steel hooks, as indicated in Fig. 247, or

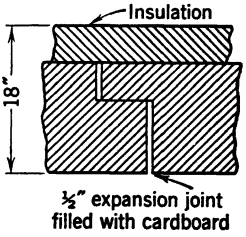


Fig. 246. Expansion joint.

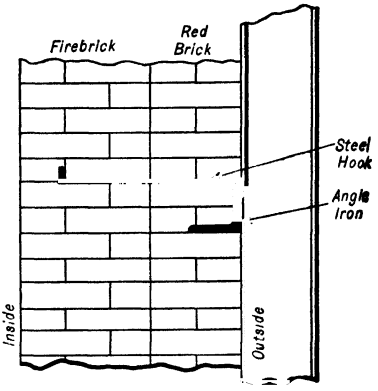


Fig. 247. Method of supporting large walls from falling into furnace.

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* Because of the difference in size, not all bricks can be bonded, unless inordinately thick layers of mortar are used. Fireclay bricks, silica bricks, silicon carbide bricks, and insulating bricks have dimensions of 9 by 4½ by 2½ in., or else 9 by 4½ by 3 in., whereas the dimensions of common red bricks are from 8¼ by 4 by 2½ in. to 8½ by 4½ by 2½ in. The size of pressed bricks is different, namely, 8⅜ by 4 by 2⅜ in.

by flat strips of metal joining the different parts of the wall. The use of hooks or strips requires costly chipping, which can be avoided by the use of standard bricks with T-shaped recesses. It should be remarked here that sidewalls of the design shown in Fig. 248 (the reasons for which are explained below) are not held at the top and must be considered as being supported from the bottom only.

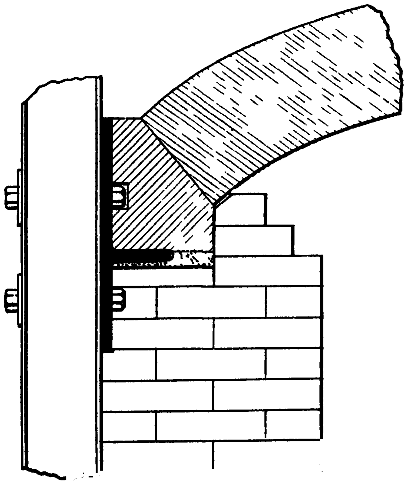


Fig. 248. Furnace arch supported from the binding.

Modern furnaces, built of insulating bricks, are protected on the outside with steel plates, because these refractories are soft and brittle in varying degrees, and because blanket, or block insulations are, as a rule, used in thin layers on the outside (frequently 1 or 2 in. thick) Attention may again be called to the fact that most insulating materials disintegrate with age.

As a rule, sidewalls are held at the top against falling inward, because the thrust of the arch presses the skewback against the binding and, by virtue of friction between the skewback and the top of the wall, holds the wall in position. But arches of wide span, acting as roofs in 2200 F furnaces, are liable to collapse if there is the slightest yielding of the abutments, such as might occur by sagging of the sidewalls. It is, therefore, frequently preferable not to rest such arches on the sidewalls, but to support the skewbacks directly from the binding, as shown in Fig. 248. This construction is safer for the arch, because the arch is not affected by the heat expansion or sagging of the sidewalls; but it is worse for the sidewalls, because their tops are no longer protected from falling inward. The top of the wall moves inward, because the hotface bricks near the bottom, where the overlying load is greatest, yield slowly, while the outside (cold face) bricks retain their shape. Corbeling the top of the wall back, as shown in Fig. 248, has been suggested as a remedy against its falling inward, but is of no avail. Tying the wall, as shown in Fig. 247, is a better remedy, but the best insurance is the use of high-grade refractories, which do not yield under the combination of temperature and pressure. It should be mentioned that the steel work which supports the roof is protected from heat by brickwork and by sand in the expansion space.

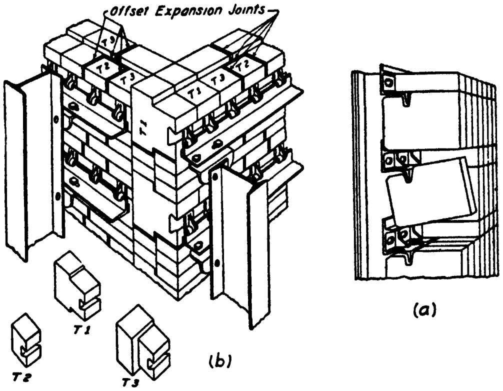


Fig. 249. Two types of metal-supported sidewalls.

A rammed plastic mix is even better than corbeled brickwork on top of the wall. Suspended roofs support sidewalls at their tops.

Metal-supported walls have long been used for large steam generators. In that use, the refractories are subjected not only to high temperatures but also to the slagging influence of flying ash from powdered coal. Although this combination is extremely rare in industrial furnaces, large industrial furnaces are occasionally built with externally supported walls. In such a wall, the tiles do not rest on each other; each tile is separately hung from a metallic support. Two designs are shown in Fig. 249. These walls have the advantage that each tile can be replaced without tearing out other tiles; that spalling is reduced on account of freedom for expansion; and that the lowest tiles are not subjected to heavy superimposed loads. On the other hand, it is questionable whether the joints can be as tight against leakage as in the ordinary wall.

The supported wall, and especially the supported nose (see Fig. 219) where a vertical wall joins a lowered section of the roof, have made possible many designs which, although theoretically correct, were formerly ruled out by the limitations of construction details. A design with two supported noses is shown in Fig. 250.

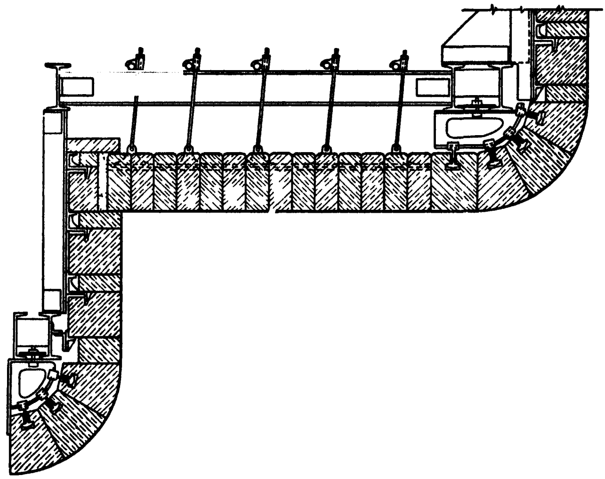


Fig. 250. Construction of sidewall between nose and roof. (Bigelow-Liptak Corporation.)

Openings in the sidewalls can be a prolific source of trouble in furnaces that carry from 2100 to 2300 F temperatures, particularly if these holes are flue openings. Figure 251 illustrates this case. If the furnace is driven hard, the outgoing products enter the flue with a temperature that may reach 2450 F. If secondary combustion (brought about by mixing of gas and air) takes place, the temperature can reach 2550 F. The tile 1, which spans the flue receives heat from three sides and assumes a temperature considerably in excess of the nominal furnace temperature. The creep strength of ordinary firebricks is extremely low at the above mentioned temperatures. In consequence, the tile sags under its own weight and that of part of the overlying wall,

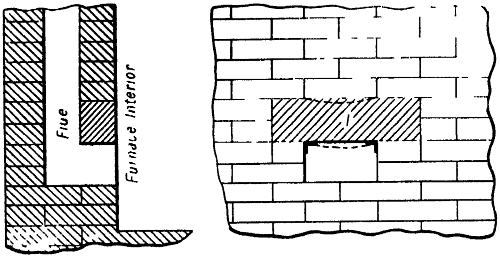


Fig. 251. Sagging of flue tile caused by overheating.

and the flue opening is gradually closed up. Evidently, only the best bricks, such as high-alumina bricks or superduty bricks, should be used in such a location. Silica brick is frequently suggested for cases of this sort, but every attempt to use it results in failure. The reason for the failure is the sudden expansion of silica brick in the range below 1000 F, as previously explained. This expansion also prevents the bonding of silica bricks with firebricks, because they will crack off sharply at the first heating up. A silicon carbide tile will be satisfactory if the flue gases carry no liquid or sticky iron oxide.

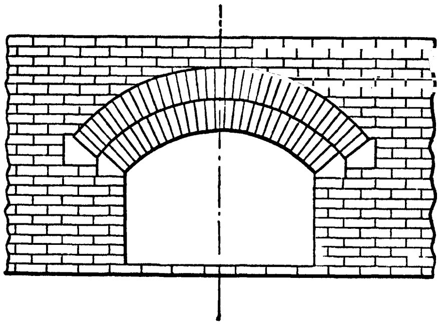


Fig. 252. Double arch over flue.

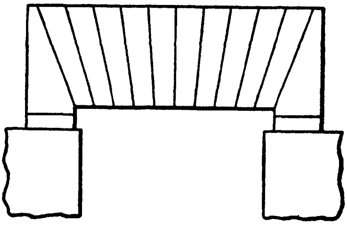


Fig. 253 Deep jack arch, flat top and bottom.

Tension in the tile, 1, of Fig. 251 can be avoided if the tile is replaced by an arch, but even an arch will sag under the effects of heat from many sides and under a superimposed load, unless the very best firebricks are used and laid in high-grade furnace cement.

An improvement in design, consisting of a double arch, is shown in Fig. 252. The upper arch carries the load; the lower arch carries only its own weight, but protects the bottom of the upper arch from becoming overheated. Results in practice have been very satisfactory. A deep jack arch, such as shown in Fig. 253, serves a similar purpose. The bottom ends of the bricks may be softened by the heat, but the top portion sustains the load.

Other troublesome openings in sidewalls are door openings, the arches of which occasionally collapse and fall into the furnace. Quite often the collapse is due to mechanical abuse of the furnace, such as striking of the door jambs by ingots, blooms, or billets which are being charged. Occasionally, the door is pulled away from the furnace with the charge (while the latter is being removed) and then allowed to slam back against the furnace. But, aside from such abuse, door arches fall in, not from any flaw in themselves, but on account of the giving way of their supports. This action is illustrated by Fig. 254. It occurs in its most pronounced form in piers between two doors in forge and heating furnaces, and particularly if the flame or heat is thrown toward the door to heat the stock near the door. Flames envelop the pier at least on three sides, and sometimes on the outside, because of poorly fitting doors. The pier assumes a high temperature through most of its cross section. The temperature is highest at the inner face and at the corners (1, see Fig. 254). Piers carry higher loads per square inch than sidewalls. The combination of high temperature and of high pressure causes the bricks near the bottom to yield. Although the yielding of the inner face is slow, it finally causes the pier to incline inward, tips the door arches, and allows them to fall into the furnace.

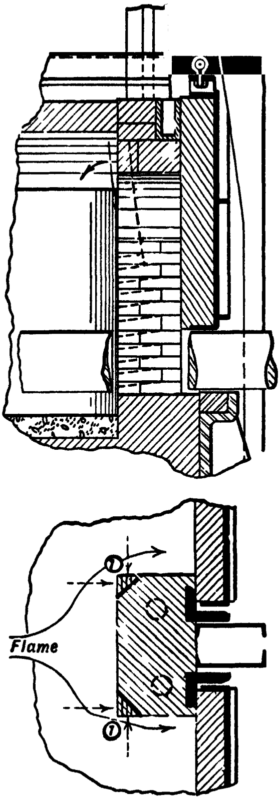


Fig. 254. Action of heat on piers.

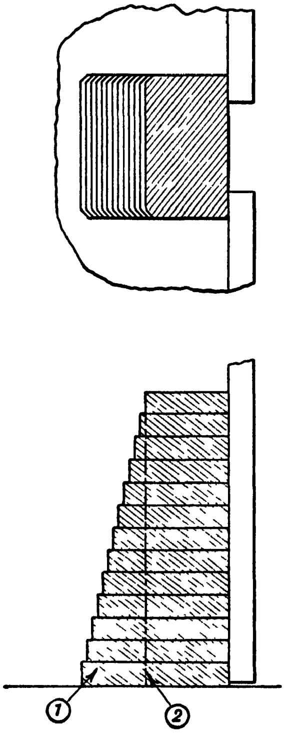
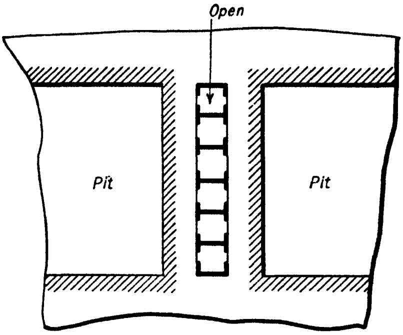


Fig. 255. Pier with thickness increasing from top to bottom.

Several solutions exist for the problem of making the piers more durable. High-alumina bricks or superduty bricks withstand the combined actions of temperature and pressure much better than ordinary firebricks. The use of high-grade refractories is indicated for piers which are exposed to severe conditions. The door jambs are often rounded off, thus doing away with the corners, 1, which would soften in any event. The pier can be made thicker at the bottom than at the top, which means that at the bottom of the pier, where the load is greatest, those bricks that carry the load are protected from the greatest heat. This is shown in Fig. 255. Bricks at 1, which are at high temperature, carry no load but protect the bricks at 2, on which the weight of the wall rests, from becoming overheated. Finally, as a last (but not recommended) resort, pipes can be arranged to circulate cooling water, as indicated by the dotted circles of the plan view of Fig. 254. According to a "rule of thumb," the width of the pier between two doors should not be less than 24 in.

Fig. 256. Air-cooled partition wall in a pit furnace.



Partition walls and bridge walls, which are heated from two sides, must be built of high-grade materials if subjected to temperatures in excess of 2300 F. In early designs, the partition walls of ingot-heating furnaces, which must withstand damage from leaning ingots in addition to the effects of high temperatures, were often cooled by the principle shown in Fig. 256. However, in all modern furnaces of this type, the preferred design is a solid wall of 40 in. minimum thickness. Such walls are constructed of high-grade firebrick, rammed plastic refractory, or silica stone, and give excellent service under these severe conditions if reasonable care is exercised in heating and cooling of the furnaces. Most of these partition walls are in excess of 20 ft in length, and the longer walls are frequently built as shown in Fig. 257.

A very difficult partition is encountered in zone-partitioned doughnut furnaces (rotating-hearth furnaces). Such partitions are shown in elevation on the left-hand side of Fig. 133. Maintaining these partitions in large furnaces is annoying and expensive, because the inner and outer circular walls expand non-uniformly and at different rates

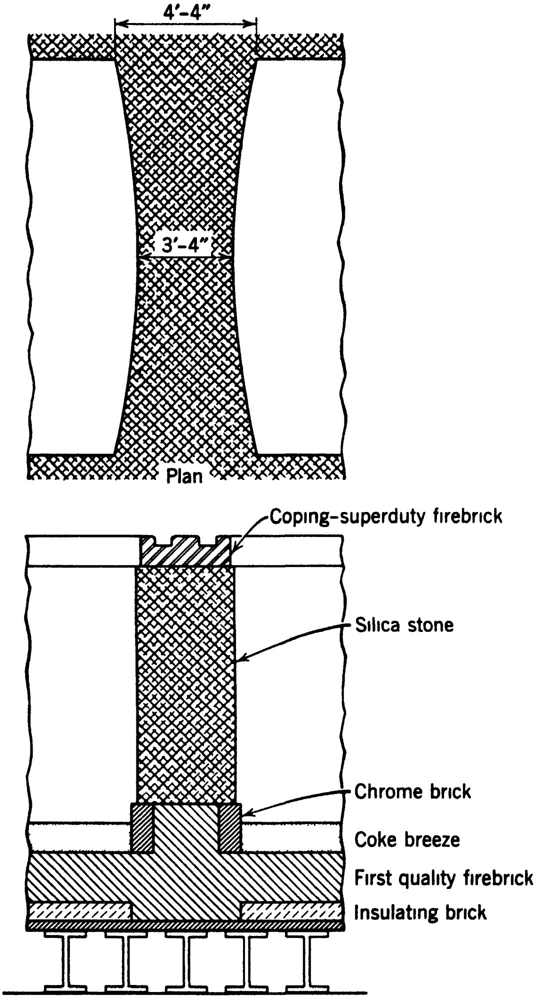


Fig. 257. Partition wall in ingot-heating furnaces.

around the circumference. The partitions, which in this case are called baffles, rest either on arches (as in Fig. 133) or else on water-cooled pipes. Even then the baffles give way; in the hot zone, the supporting arches sag. In order to avoid this trouble, several superimposed arches are sometimes installed, in conformity with the principle illustrated in Fig. 252. In spite of the use of these expedients, the life of the baffles is short. A recent design utilizes chain walls, constructed as shown on Fig. 258. The trouble with baffles has been eliminated in Europe by substituting a knuckle in the roof for a baffle and arranging burners in the roof of the knuckle, close to the inner and outer diameters of the circular hearth. Another European design comprises a double wall with an air-cooled baffle in between.

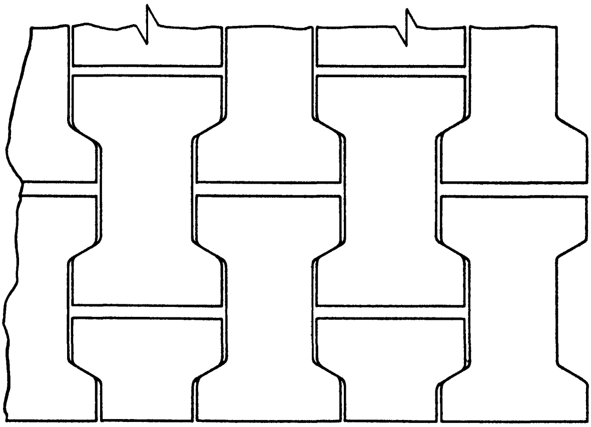


Fig. 258. Partition wall constructed of interlocking refractory shapes.

Bridge walls are special partition walls, over which flame or flue gases flow from one chamber to another. They are found in regenerative furnaces and in so-called direct-fired furnaces in which coal is burned on the grate. They are also used in furnaces that are fired with powdered coal. They are quite hard to maintain for the reasons that were expounded for partition walls, but present the added difficulty that ash particles, flame, or iron oxide impinge directly on them. No grade of firebrick withstands the combined influence of these actions. The result is that bridge walls must be repaired quite often; in many plants they are repaired every Sunday. Cooling of the bridge walls helps considerably. Figure 259 shows how secondary air is carried through the bridge wall of a coal fired furnace. This arrangement furnishes air for final combustion at the right place and cools the bridge wall. The same illustration incidentally shows a ventilating duct under the bridge wall for the purpose of keeping the slag from breaking through into the ashpit, which is at times quite hot from accumulated red-hot ashes. Another form of ventilated bridge wall which is very common in regenerative furnaces is shown in Fig. 260. Incidentally, this illustration also shows a ventilated hearth.

As far as the mechanical strength of the bridge walls is concerned, it must be remembered that they have no lateral support at the top. For this reason, that part which is exposed to high temperatures on both sides should be made not less than two-thirds as wide as it is high, if the furnace temperature exceeds 2200 F. It is of assistance if the bridge wall is made wider at the bottom than at the top. With lower temperatures, bridge walls can be somewhat narrower.

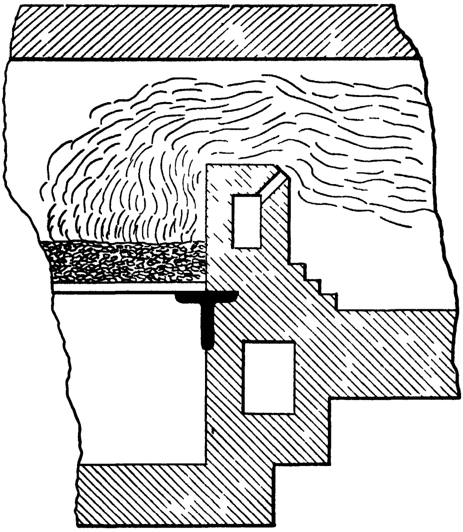


Fig. 259. Ventilated bridge wall carrying secondary air

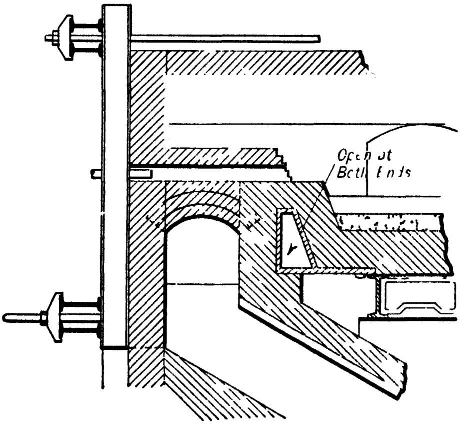


Fig. 260. Air-cooled bridge wall in a regenerative furnace.

6. PROTECTION OF WALLS AND ROOFS BY BONDING MORTARS AND CEMENTS

Bricks are irregular in shape and, after having served in the furnace for some time, often present an appearance similar to Fig. 261, which is exaggerated for the sake of clarity. Corners of bricks are exposed on two or three sides to heat, ashes, iron oxide dust, and other destroying influences. The corners melt (by excessive temperature) or spall off (by rapid heating), exposing other surface, particularly if furnace temperatures of 2400 or 2500 F are carried for the purpose of rapid heating. To make up for the irregularities in the shapes of bricks, they are laid in mortar. Since bricks are made of fireclay, and since their particles are bonded together very strongly, it might, at first thought, be expected that fireclay mortar bonds bricks together with an equally strong force. The bonding power of fireclay mortar is, however, very much smaller. The bricks were machine-pressed and were then burned at a very high temperature. Obviously, the mortar has received no such treatment and has, in consequence, much less bonding or cohesive force than the brick particles. The bonding power depends, to a large extent, on the composition and fineness of grinding of the dry clay. A cheap mortar either melts and runs out or else acts as a dry filler only, which falls out. In either case, the condition depicted in Fig. 261 is produced. Mortars of either one of these two descriptions are harmful fillers in places that are exposed to high temperature and pressure as, for instance, on the furnace side of arches or at the bottom of tall walls. The harmful effects of fireclay mortars are not appreciated

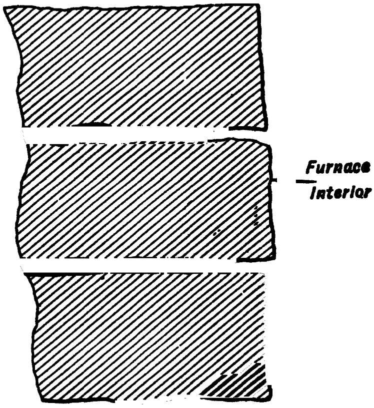


Fig. 261. Exposure of edges and corners to furnace temperature after crumbling of fireclay mortar.

by bricklayers who have worked in the building trade (on the wall) and who make the joints much thicker than necessary.

In places that are exposed to high temperature and pressure caused by superimposed loads, it is advisable to rub the dry bricks together to grind off irregularities, or else to use a high-grade mortar in a very thin layer.

Bonding mortars* have been improved, and excellent mortars are now available for each type of refractory. Mortars of two types are on the market. They are: (1) hot-setting mortars, and (2) air-setting mortars. Hot-setting mortars do not acquire the property of bonding until a certain temperature has been reached. If, for instance, the mortar consists of first-quality fireclay with a fusion temperature of 3000 F, then the mortar is a filler without bonding power in a furnace whose wall temperature does not exceed 2200 F. And even if the furnace were temporarily brought up to a temperature of 3000 F, to make the walls monolithic, the bonding temperature would be reached only on the furnace side of the walls, but not from there out. In order to make the mortar bond for some distance from the inside of the furnace, the temperature would have to be run up much beyond the bonding temperature. A mixture of about 10 per cent of salt, lime, or Portland cement (or similar ingredients) lowers the fusion point (and with it the bonding temperature) about 500 F.

On account of these properties of hot-setting mortars, cold-setting or (to use the common name) air-setting mortars are frequently preferred. For use in industrial furnaces, such mortars consist either of a high-quality fireclay, or of silica, to either of which a binder has been added. The makers of air-setting mortars keep the nature of their binders rather secret; it is, however, known that sodium silicate is a commonly used constituent. Since the binder reduces the fusion temperature of the mortar, the base material of the mortar must have a very high fusion temperature. Such base materials are: high-alumina (diaspore) clay, quartzite, and chrome ore. Additional base materials for bonding mortars are in existence; however, they form mortars for bricks that are not used in industrial furnaces.

Air-setting mortars bond all the way through the wall, from the hot surface to the cold surface. In order to bond the mortar to the bricks, the binder in the mortar must chemically combine with the material of the bricks. Time is required for this action, and the fusion temperature of the bricks is lowered at the interfaces between bricks and mortar. For that reason, commercial mortars contain no more binder than is absolutely necessary. Because air-setting mortars require time for proper binding, quick furnace repairs are often made with hot-setting mortars. Special bonding mortars have been developed for insulating bricks.

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* In former years, the term "furnace cement" was more commonly used than the term "bonding mortar." When a reclassification of goods made the freight rates on cements higher than those on mortars, most furnace cements became bonding mortars. Although the freight rates are now the same, the term bonding mortars is now most commonly used.

Obviously, the properties of bonding mortars concern mainly the builders of industrial furnaces. But they also concern the owners of furnaces, because furnaces do not last forever and must be repaired from time to time, or even be rebuilt.

In the present-day language, the term, "furnace cement" is limited to materials that are either brushed on or sprayed against the inner face of furnace walls or roofs. Industrial furnaces that are shut down over the week end, and are then heated quickly on Monday morning, suffer from spalling. Portions of the bricks drop off or out here and there, and the once smooth surface becomes extremely rough. Although this undesirable condition might be avoided by slower cooling and heating, it is often preferred to recondition the furnace interior by filling the holes. A spray gun is the usual tool, because holes are often so deep that brush coating is out of the question. The furnace cements usually consist of ground fireclay and bonding mortar. The latter is adapted to the temperature that ordinarily prevails in the furnace.

The suitability of furnace cements is best determined by trial. A common method of comparing furnace cements is to try two brands on different but equally difficult parts of the same furnace. Too much must not be expected of brush or spray coatings. They last a certain time and then gradually crumble away. In spite of this fact, repeated building up of furnace interiors by spraying is usually cheaper than rebuilding the furnaces. Brush coatings are often applied to new furnaces to hasten their becoming monolithic.

Although coatings on the outside of furnace walls are for heat economy rather than for strength and durability, coatings for the outside of furnace walls are discussed here. Such coatings are applied to furnace walls and checkers to prevent air infiltration and reducing heat losses. Mixtures of clay and sand or diatomaceous powder, with molasses or similar material as a binder, have been used. The coating is applied by trowel. A better material is fibrous asbestos with bitumen as a binder. Gas-producer tar is often used as coating, being thinned before application with gasoline or other solvent. Insulating cements which withstand temperatures up to 2800 F and which adhere well to walls are now on the market.

7. DOORS

Furnace doors should be tight, light, durable and heatproof. In this division, the problem of obtaining as many as possible of these desirable features will be taken up, especially that of securing strength and durability.

By far the greatest number of doors are of the lifting type. Hinged doors have two disadvantages. One is that, with hinged doors, it is impossible to heat a piece which is to be partly within the furnace and partly out of it. The second objection is that hinged doors are, as a rule, slammed in closing; this loosens the refractory material and causes it to drop out. In small furnaces, such as tool-hardening furnaces, for instance, these objections are not valid enough to forbid the use of hinged doors, and designs similar to that shown in Fig. 262 are used. A single tile is held firmly against the door frame by one or more bolts, the ends of which are protected against the heat by fireclay. If the tile breaks, a new one is easily inserted.

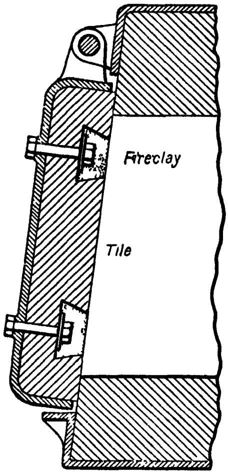


Fig. 262. Hinged door for small furnaces.

Large hinged doors serve well at the discharge end of continuous slab-heating furnaces. A swinging door for such a furnace is illustrated in Fig. 263. The door consists of one or more castings; if more than one, they are bolted together. It is lined with firebrick or castable. A door of this design could not long withstand the repeated impacts of slabs or blooms sliding down the incline. For that reason, the door-opening mechanism and the pusher mechanism are electrically interlocked in such a manner that the pusher mechanism cannot operate until the door is wide open. Part of the door-operating mechanism is shown in Fig. 263. The rest consists of an electric motor, a gearbox, a solenoid brake, and additional levers, links, and shafts, the arrangement of which varies with the available space.

The metal frames of lifting doors must be strong and rigid enough to carry the refractory lining without danger of cracking it, and to withstand the tendency to warp which is caused by the heat expansion of the lining. To minimize heat losses, the refractory lining should be both thick and of good insulating quality; yet the weight of the door should be small, so as to offer little frictional and inertia resistance to opening and closing. Furthermore, doors must withstand the action of heat escaping from the furnace.

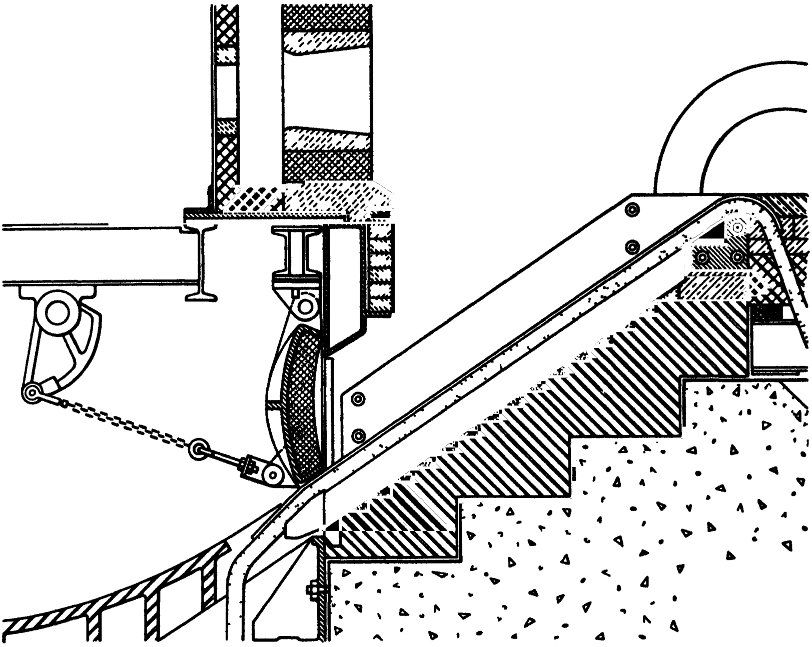


Fig. 263. Mechanically operated door at discharge end of continuous furnace.

If the furnace temperature does not exceed 1600 F, the design and maintenance of doors are comparatively easy. The low temperature permits the door lining to be made of a highly insulating material, such as the plastic insulating materials which have been previously discussed in this chapter. These can be rammed into the door frame to form a monolithic lining. The weight of such a lining is between

50 and 60 pcf, and the thermal conductivity is from one-fourth to one-third that of dense firebrick.

Small doors of less than 4 sq ft in area do not require reinforcing, but they should have the edges protected and be not less than 2½ in. thick. Figure 264 shows how the cement is held against the door frame by hooks or anchor bolts. If of cast iron (as is usual), the door frame is made just as thin as can be cast, which means ⅜ in. for doors of up to 8 or 9 sq ft, and correspondingly more for larger doors.

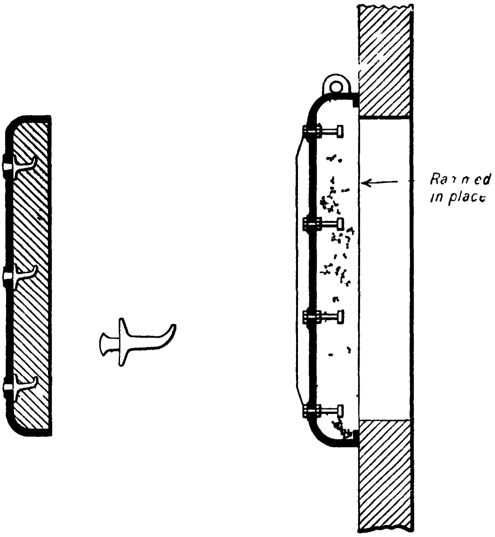


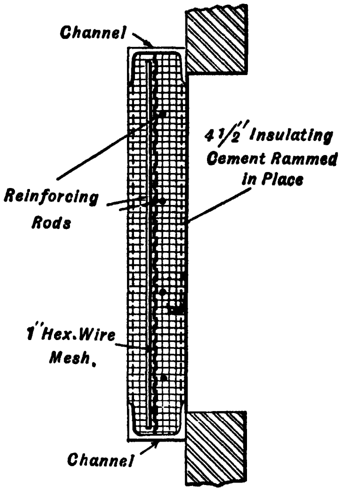
Fig. 264. Methods of holding cement in door frame

Large doors which contain a rammed mixture require reinforcing, either by ribs in the door frame or else by running tierods through the door frame vertically and horizontally, and placing between the two sets of rods wire netting of ½- to 1-in. mesh. If the reinforcing is placed in the middle of the 4½-in. thickness of cement, no backing is required, a steel channel frame with the before-mentioned tierods being sufficient to give strength to the door. Steel channel frames, however, should be confined to furnaces at low temperatures, or to those which are heated by resistance elements or by radient tubes, where the furnace is not under pressure. A design of this kind is shown in Fig. 265. Wherever it is desirable to project the lining past the door frame, it is advisable to run a course of brick on edge completely around the door frame, setting the brick out to the desired projection. The insulating cement is then finished up flush with the brick edge, or else applied so that the brick projects a quarter of an inch.

Monolithic door linings which will withstand temperatures up to 1900 F consist of crushed insulating firebricks with calcium aluminate (such as trade named Lumnite) as a binder. The mixture can be rammed into place but is more conveniently cast in place.

The design and maintenance of doors are easy so long as the furnace temperatures do not exceed 1600 F, but difficulties arise as the temperature grows higher. The doors and door frames of certain 2200-F furnaces cause weary hours to both furnace builders and furnace operators. The doors consist of frames which are lined with firebricks which become loose, catch on the furnace front, and drop out. (Laying the bricks in good bonding mortar helps to avoid this.) In addition, linings suffer from mechanical abuse. Taking material out of a hot furnace when the door is wide open is a disagreeable task. The door is lifted no more than is absolutely necessary, and the workmen, in their hurry to remove the stock from the furnace, catch the bottom of the door, pull it away from the furnace, and then drop it back against the seat. Very few door linings withstand this treatment for any length of time. This applies particularly to the ordinary designs of door frames, such as Figs. 266 and 267, which are green sand castings made without cores. Solid doors of cast iron (Fig. 266) are common for door openings up to 2½-ft width. For wider openings the iron work is open, as in Fig. 267.

Fig. 265. Furnace door with reinforced insulating cement.



Undercut doors, such as shown in Fig. 268 (which represents horizontal section through such doors), retain the brick lining much better, but they are more expensive in the foundry. Evidently, the retaining frictional force in Fig. 268 depends on the care and force with which

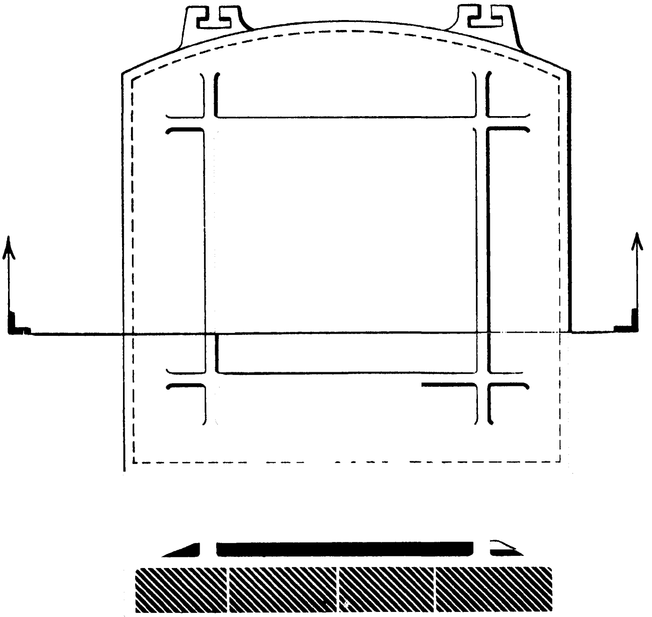


Fig. 266. Common design of furnace door, solid type.

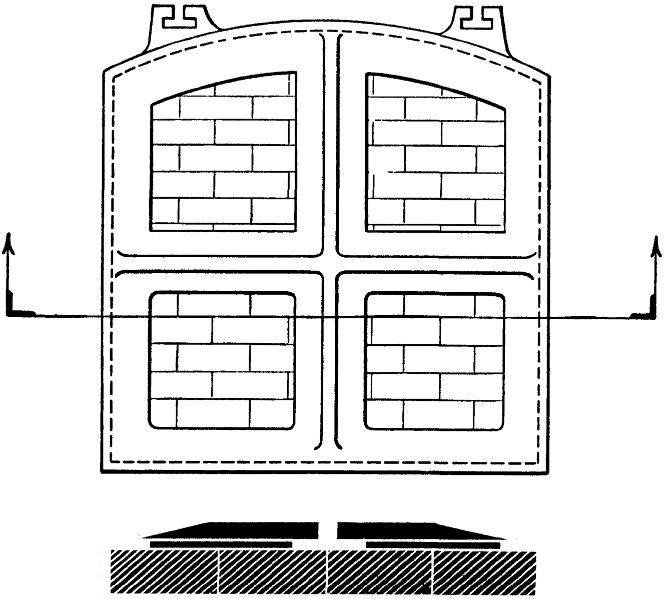


Fig. 267. Common design of furnace door, open type.

the bricks are pushed into place, and also on the area of the door. Quite obviously, the binding force of the inclined edges does not extend to the center of large doors. The use of high-temperature bonding mortars for door lining is recommended, both for the first laying of the bricks and for week-end repairs.

A very rough-looking but very substantial solid door is obtained by setting firebricks into a mold and pouring cast iron over and around the bricks after they have been heated. The iron penetrates into the cracks and, in shrinking, grips the bricks so tightly that they are held permanently. Only spalling or burning off of the bricks can put this door out of commission. The only way to repair such a door is to make a new one.

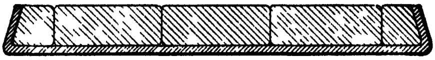


Fig. 268. Furnace door with undercut frame.

Doors are made to be opened and closed. When the door is open, the lining is cooled; and when the door is closed, the lining is heated again. This alternate change of temperature produces spalling in refractory bricks of many types. It is claimed that certain refractory cements withstand temperature changes very well.

For the sake of lightness, the firebrick lining of doors is frequently made only 4½ in. thick. If dense firebrick constitutes the lining, the heat loss through the door is very great. Insulating firebrick is much to be preferred as a lining. Even if made 9 in. thick, insulating firebrick do not make a heavy door.

Doors exceeding 36 sq ft in area are very heavy if the frames are made of cast iron. Furthermore, large cast doors are unreliable on account of casting strains. In addition, it is very hard to hold the bricks in a large plain door. Finally, warping of large doors cannot be prevented, unless the metallic backing of the refractory lining is very rigid. Sheet steel to which strong I-beams and channels are welded meets this specification. A rigid door of this description is illustrated by Fig. 269. The refractory lining is fastened to the metallic backing by any of the above-described methods, for instance, by T-shaped hooks that fit into recesses in the bricks. Inspection of Fig. 269 reveals the fact that a limited number of bricks is thus fastened. The rest are held in place by friction.

Figure 270 is a view of part of a car-type furnace. When serving for stress-relieving large tanks, such furnaces assume huge dimensions. Vertically moving doors would require either a tall tower for guiding

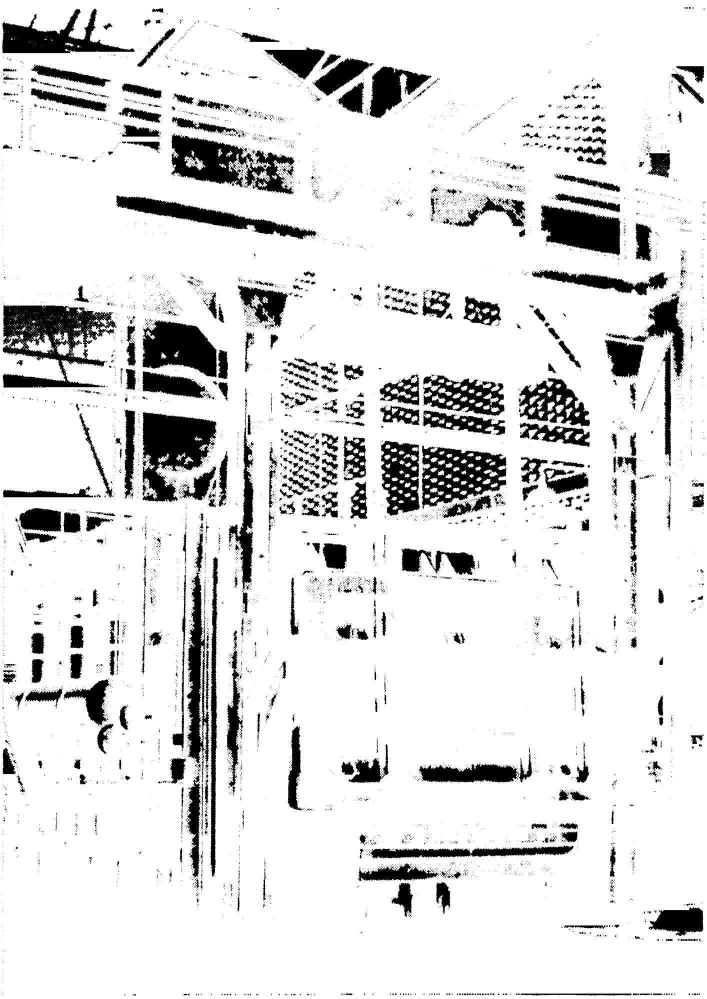


Fig. 269. Rigid door of structural members. (Courtesy of Wilhelm Ruppmann)

or else a deep pit, either of which is objectionable. The difficulty is overcome by making the door an integral part of the car. The structural work by means of which rigidity is obtained is well shown in the illustration.

For the benefit of furnace operators, water-cooled doors are occasionally used, although in heating furnaces, strength and durability do not require their use. Where a number of doors are located along the front of a furnace, water-cooled front skewbacks greatly lengthen the life of arches which are sprung from front to back of such furnaces.

Practically all vertically lifting doors are balanced by counterweights (Fig. 271). The latter reduce the work of the door-lifting mechanism and assist in keeping the door open in any desired position. Some engineers have made the counterweights adjustable for the purpose of maintaining perfect balance when part of the lining has dropped out. The loose parts of the counterweights are, as a rule, mislaid and get lost. Large heavy doors are operated by power (electric, hydraulic, or pneumatic). Counterweights are not necessary with power operation, but are nevertheless installed to assist in manual door operation in case of power failure.

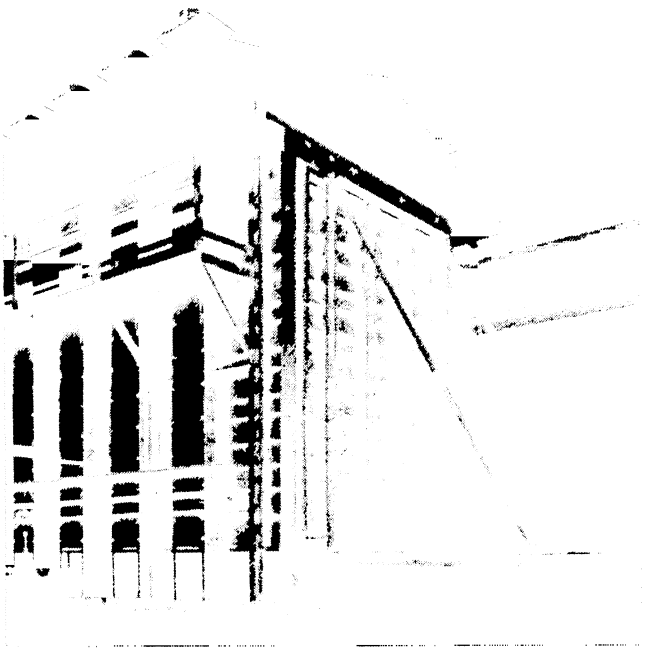


Fig. 270. Structural door mounted on furnace car.

Counterweight mechanisms should be so designed that they are not in the way of the operators, and at the same time are not exposed to the action of flame escaping from the doors. Sheaves should be of the roller-bearing type, since plain bearings under the action of heat produce such excessive friction as largely to nullify the effect of counterweighting. Wire cables operate more smoothly and with less force than chains, but the latter are frequently used for large doors at high temperatures. If the weights enter a pit in the floor, guards should be provided to protect the operator.

A part of the door that needs attention is the bottom, particularly in forge furnaces. The latter are frequently so operated that flames or hot gases pass out around the billet or ingot on which the door rests. The bottom of the door is then exposed to flame and hot products of combustion, burning off while the rest of the door frame is still in very good condition. Although these circumstances favor making the bottom part of the door removable, the problem is how to make it so without introducing new troubles. There must be no bolts on the inside of the door, because they will burn in so tightly that they cannot be removed. Bolts that are located on the outside have a short lever arm and should be assisted by bolts on the side of the door as shown in Fig. 272.

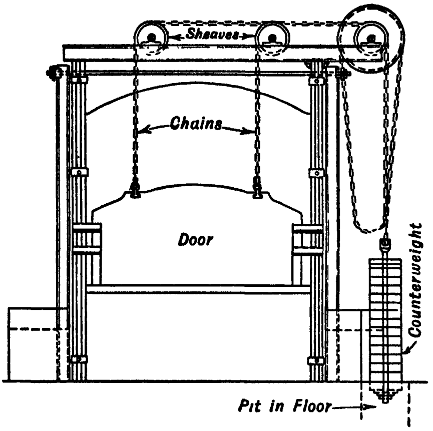
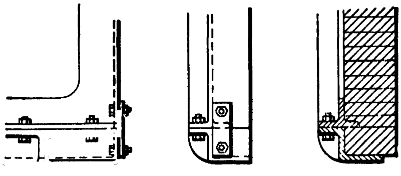


Fig. 271. Door-lifting mechanism with counterweight.

Fig. 272. Removable door bottoms.



The rib on the inside of the lower end of the main part of the door is quite important, keeping the lining intact during the replacing of the bottom.

Forge furnaces are responsible for another type of door which is quite troublesome—namely, the split door. The use of split doors is encouraged by the fact that two forgings of different sizes can be heated in one door opening if a split door is used (see Fig. 273). The split door is troublesome, however, because the proper guiding of each half is very difficult. Unless great care is used, the two parts flop around, hang out of square, allow wide openings for flame discharge and the burning out of doors, door frames and furnace fronts. The split door is a nuisance and should be avoided whenever possible. It can be improved by guiding an upward extension of each door above the door opening.

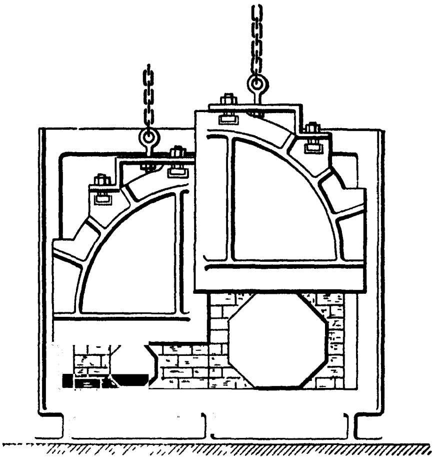


Fig. 273. Split door.

Good furnace doors fit tightly. Combustion-type furnaces are operated with a slight pressure to keep air out, and, even if that pressure is only 1/30 in. of water, loose-fitting doors allow flame and products of combustion to escape. Even if the draft should be perfectly balanced at the hearth level or if electric heating is used, the height of the heating chamber will set up a draft, allowing cold air to pass in at the bottom and heat to pass out at the top of the door if there are leaks. In this sense, tightness of doors insures greater durability.

Tightness depends partly on correct suspension of the door. On account of the burning away and spalling of the lining, it is difficult to suspend doors so that they always close tightly, unless adjustments are provided. A door that is lifted by two chains needs only an in-and-out adjustment, but a door lifted by only one chain needs also a right-and-left adjustment. Ordinarily, adjustment is obtained by branching one chain in two, or by using two links, as shown in Fig. 274. Bending of the links is used for adjustment.

Unfortunately, the adjustment is not permanent, on account of spalling or crumbling of the lining, and also because of mechanical injury. For that reason, a quicker and more certain adjustment is desirable. Such a method is shown in Fig. 275. T-headed bolts sliding in slots provide in-and-out adjustment, while an eye-bolt with a large washer, sliding in an oblong hole, provides for right-and-left adjustment. The crossbar must, of course, be made strong enough to carry the weight of the door.

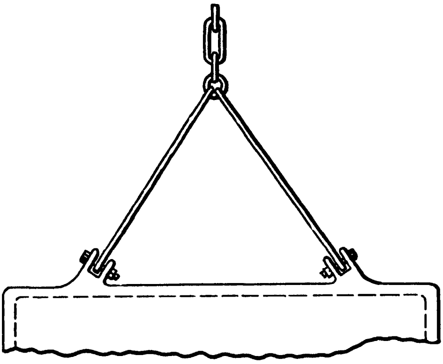


Fig. 274. Door suspension. Links can be shortened by bending.

Even with the most careful hanging of doors, it is difficult to make them close tightly, unless good door checks are provided for holding the door against the furnace front. A good and inexpensive door check is illustrated in Fig. 276. A wedge-shaped face piece on the door comes in contact with a long leaf spring, which is cut and bent to secure safe entrance, even if the door should hang away some distance from the frame. If so designed that the springy part is protected against excessive heat, a spring check of this description is much better than a solid door check. The latter is broken off, sooner or later, by abrupt lowering of the door. Besides, a door cannot bear tightly against rigid door checks and, at the same time, close tightly at the bottom unless there is a sand bottom, as in Fig. 277. With a sand bottom, the rigid door check is satisfactory. Most door checks are of

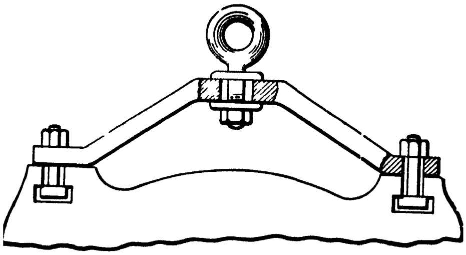


Fig. 275. Adjustable suspension of furnace door.

no use with doors that work with various heights of opening, such as doors on forge furnaces.

The door check shown in Fig. 269 is an exception. A U-shaped bar resting on two projecting, inclined arms lightly presses the door against the furnace front in any position of the door.

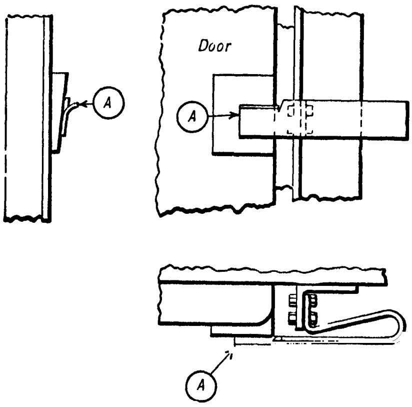


Fig. 276. Furnace door check.

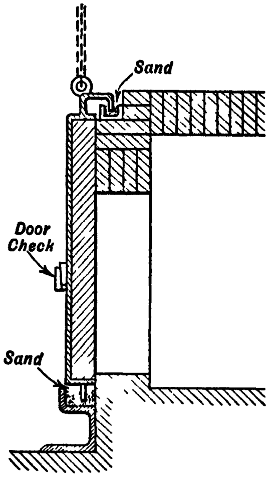


Fig. 277. Door with sand seal at top and bottom.

A door-tightening device that is operative at any height of door opening, and that cannot be broken by careless handling, is obtained by making use of gravity. To this end, doors are made slightly inclined, as in Fig. 278.

In that case, the lifting chain must pull parallel to the face of the door, and the door must slide on the furnace front. If brick slides on brick, there is constant danger of catching and ripping the linings out of the door; and if metal slides on metal, there is danger of burning these slides away, on account of the before-mentioned pressure in the furnace, or because of the chimney effect. In some furnaces, the sliding surfaces, marked by arrow 1 in Fig. 279, are protected against excessive heat by a broad surface of refractory material. Although this design is very good for protection of the frame, it exposes the refractory lining to mechanical injury. For that reason, the design of Fig. 280 is commonly adopted for door fronts, with either cast iron or cast steel as material. The sections are made heavy, and deep ribs are provided. The ribs serve a double purpose. First, they carry off

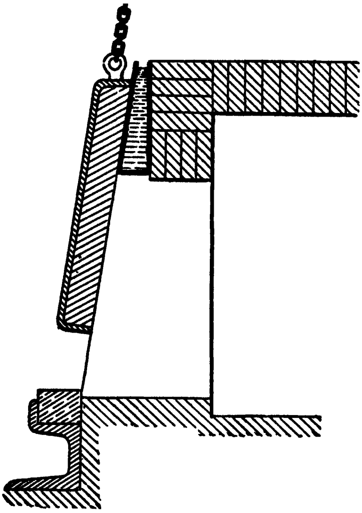


Fig. 278. Inclined furnace door.

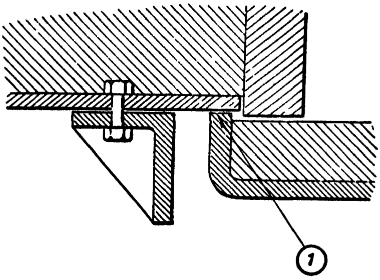


Fig. 279. Door with metal sliding on metal.

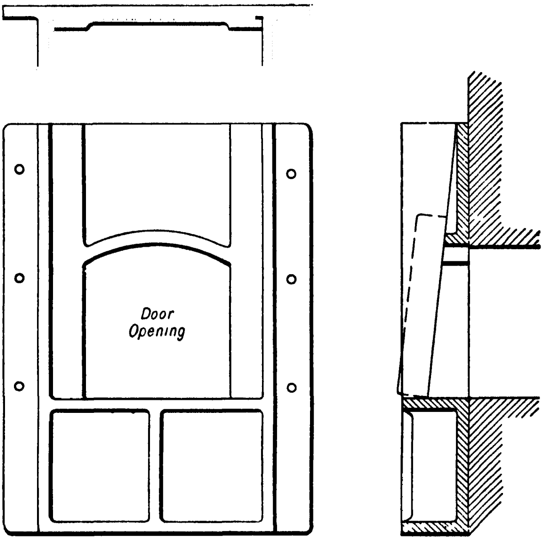


Fig. 280. Typical door frame.

and dissipate the heat which is imparted to the front at the door opening; and, second, they protect the furnace binding against impinging heat.

At the top of doors, and particularly of split doors, the burning of the door frame is frequently quite severe, although it should be understood that most of this trouble can be prevented by proper control of combustion in the furnace. Furnace engineers have tried to counteract the effect of flame at the top, either by water-cooling (Fig. 278), or by making the top part of the frame removable (Fig. 281). Quite obviously, the seal is very imperfect with doors that have to be used in a partly open position. The bolts for holding the removable top piece in Fig. 281 have been set pretty well out of the path of the flame, but even in that location they will burn in so tightly that they can be removed only by chipping. Through-bolts should be used wherever possible; if one end of the bolt is inaccessible (as in Fig. 281) it should be T-shaped and fit into a slot with lugs on the back of the casting, so that the bolt can be easily removed and a new one inserted with little trouble.

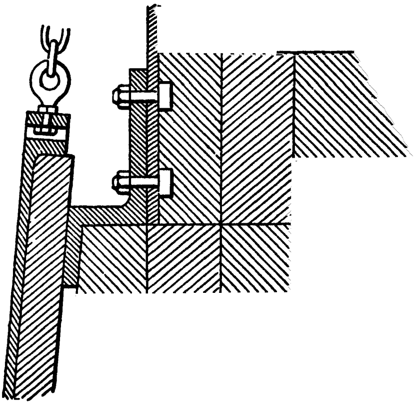


Fig. 281. Removable top of door frame.

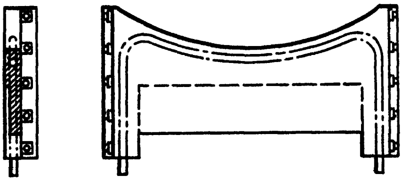


Fig. 282. Water-cooled section of door frame.

The life of door fronts is increased by casting a pipe into the front near the most exposed pieces and circulating water through it (see Fig. 282). In extreme cases, hollow door frames are used with water circulating through the whole front (see Fig. 283). Although this type of frame is quite common on open hearth furnaces, its use with heating furnaces is limited. The great temperature differences to which it is subjected make it liable to frequent cracking. This difficulty is overcome by making the front of soft welded steel.

The sills of door frames have to carry the weight of the stock while it temporarily rests on them. This fact is self-evident, and yet many broken sills are found because the draftsman overlooked this weight factor. In forging furnaces, with flame passing out between sill and door, a large billet (see Fig. 278) on the sill forms a strong, durable, and easily replaceable part.

Chain doors of two types are in use. Neither type is a true door, but rather a baffle. The chain baffle which consists of interlocking I-shaped tiles was illustrated by Fig. 258. At the charging end of continuous furnaces, refractory chain baffles restrict the flow of gases between furnace interior and atmosphere. The same purpose is served by metallic chains. As shown in Fig. 284, the chains are attached to

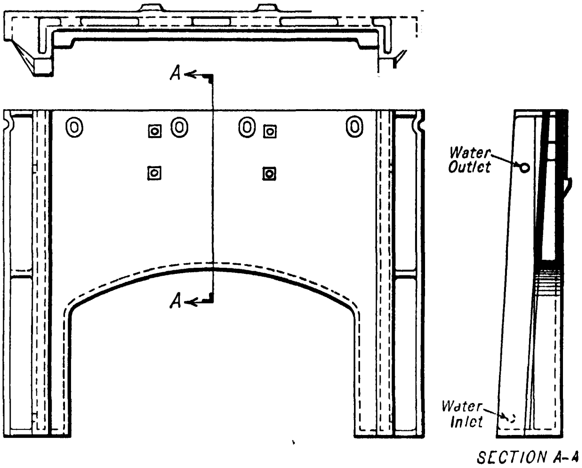


Fig. 283. Water-cooled door frame.

Fig. 284. Furnace with chain door.

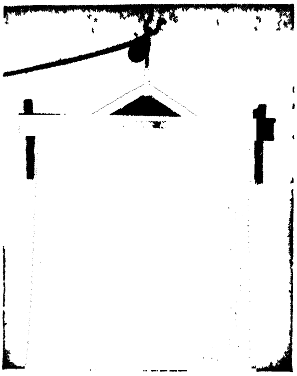


Fig. 285. Chain door. (Courtesy of E. J. Codd Company.)

the bottom of a solid door and rest on the entering stock which is to be heated. The baffle action is increased by arranging the chains in double rows, with the bottom ends made of pipes to avoid tangling. Chain doors, such as illustrated by Fig. 285, are serviceable in places where solid doors would have to be open most of the time. The operator can enter and remove stock and rearrange it through the chain door. He is protected from extreme heat, because the chains are a radiation screen.

Cast-link chains made of heat-resisting alloy have a longer life than welded chains of soft steel.

8. FURNACE BINDING

If a furnace is to stand up for any length of time, it must be bound with iron or steel (or any other strong metal). The forces to which furnace binding is subjected fall into two groups: (1) the forces that arise from the thrusts of sprung arches, and (2) the forces that arise from the expansion of straight walls, including flat suspended roofs. The methods of calculation differ in the two cases.

From Fig. 209, the forces that an arch exerts can be calculated, and magnification factors can be introduced, as explained on page 294. The arch thrust must be taken up by the binding without the yield point stress of the binding being exceeded. Buckstays and tierods or cross-beams are used, which means that the thrust of the arch must be carried by a beam between the buckstays. An example will make the method of calculation clear.

EXAMPLE. Let the arch be 9 in. thick, 10 ft span, radius 10 ft (see Fig. 286), and let the buckstays be 3 ft apart.

Distance, center line to center line of upper and lower tierods, is 9 ft 6 in. Distance from center line of upper tierod to line of action of horizontal arch thrust is 12 in.

Weight of arch section 3 ft wide $= 3 \text{ ft} \times \frac{9}{12} \text{ ft} \times \left(2\pi \times 10\tfrac{3}{8} \text{ ft radius} \times \frac{60}{360}\right) \times 125 \text{ pcf} = 3060 \text{ lb}$. Magnification factor = 2½ (page 295).

Reaction F (Fig. 286) $= \sqrt{3} \times (\tfrac{1}{2} \times 3060) \times \text{factor } 2\tfrac{1}{2} = 6600 \text{ lb}$.

A stress of 12,000 psi may be allowed in the tierods. The area of each rod must then be

$$\frac{6600 \times (114 \text{ in.} - 12 \text{ in.})}{12{,}000 \times 114 \text{ in.}} = 0.50 \text{ sq in.}$$

at the root of the thread.

The nearest size is 1 in. diameter rod.

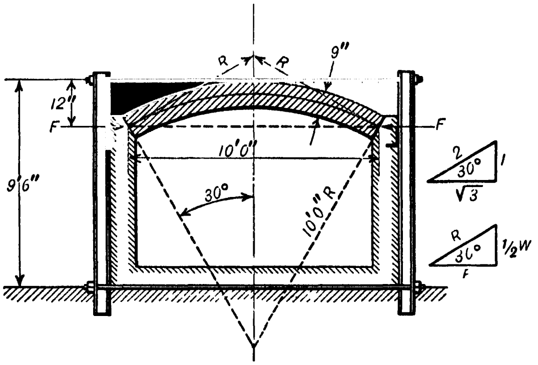


Fig. 286. Forces in furnace binding.

Bending moment in buckstays

$$= 6600 \text{ lb} \times \frac{12 \text{ in.} \times (114 \text{ in.} - 12 \text{ in.})}{114 \text{ in.}} = 71{,}200 \text{ lb in.}$$

$$\text{Required section modulus} = \frac{71{,}000}{12{,}000 \text{ psi stress}} = 5.9 \text{ cu in.}$$

From table of properties of structural shapes, each buckstay would consist of two 5-in. channels. Since the buckstays are usually cooler than the tierods, a somewhat higher stress could be permitted, and 4-in. channels could be used, if saving of weight is an object. The angle or channel which supports the arch abutments between the buckstays is a continuous beam with numerous supports. However, for reasons of simplicity and additional factor of safety, it is figured as a number of shorter separate beams, supported at the ends and uniformly loaded.

Bending moment in horizontal plane, due to arch thrust

$$= \frac{6600 \text{ lb} \times 36 \text{ in.}}{8} = 29{,}800 \text{ lb in.}$$

Bending moment in vertical plane

$$= \frac{3060 \times 36}{2 \times 8} = 6900 \text{ lb in.}$$

(if weight of arch is supported entirely by steelwork and none of it rests on the sidewalls).

A 4 by 4 by ⅜ in. angle may be used.

$$\text{Tensile stress due to horizontal B.M.} = \frac{29{,}800}{3.86} = 7710$$

$$\text{Tensile stress due to vertical B.M.} = \frac{6900}{3.86} = 1790$$

Greatest combined tensile stress, at outer lower corner

$$= 7710 + 1790 = 9500 \text{ psi}$$

The cross-binding may be either rigid or yielding. In the present example, the calculation is made for spring-loaded tierods.

Allowing a total variation of tierod tension from cold to hot of, say, 25 per cent of the initial tension when the arch is cold, a change of spring force of $0.25 \times 6600 \times \frac{114 - 12}{114} = 1500$ lb would be required. If the arch does not rise, horizontal expansion 0.075 in. per ft × 10 ft = ¾ in. Spring scale $= \frac{1500}{3/4} = 2000$ lb per in.

A spring must be selected that will sustain a load of 6000 lb when the arch is cold, with deflection of 1 in. by 2000 lb load. From spring tables, this corresponds to a spring of 6 in. outside diameter of coil, 1⅛ in. wire, 11 coils.

If 50 per cent variation of force is allowed, a spring of 4½ in. outside diameter, 1 in. wire, 10 coils, would be suitable.

Unless the arch is sprung in all directions (turtleback arch), the longitudinal binding, which is diagrammatically illustrated in Fig. 287, must be calculated as resisting the direct expansion of the walls and of the roof. The forces which affect the binding of furnaces with suspended roofs and also the longitudinal binding of furnaces with sprung arches are very much less definite than the forces that the cross binding of furnaces with sprung arches must withstand. The cross binding must not be strained beyond the yield-point stress, or else the arch will collapse. The other two bindings are often strained beyond the yield-point stress, and no harm results.

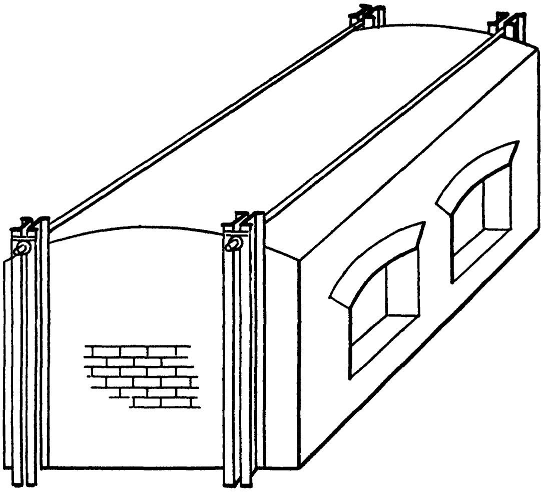


Fig. 287. Diagrammatic view of longitudinal furnace binding.

Although the hot portions of a furnace wall tend to expand more than the cooler sections near the outer face, it is permissible to figure with an average expansion, because the brick particles having different temperatures are tied together by shear stresses which tend to equalize the expansion.

At hot face temperatures between 2000 and 2400 F, firebrick has expanded about 0.075 in. per ft of length. At cold face temperatures of 400 to 500 F, it has expanded about 0.02 in. per ft of length. The average expansion is about 0.05 in. per ft of length, which equals 4/1000 of the length. Steel is stretched to the yield-point stress by a unit elongation of 1/1000. The difference is so great that any elastic compression of the brickwork cannot reduce it substantially. This means that the binding, which remains relatively cool, is stretched beyond the yield-point stress, unless it is so strong that it can suppress almost three-quarters of the expansion of the brickwork, or, more probably, unless allowance for brickwork expansion is provided, either by accident (soft fireclay between every brick) or by design.

From the few data that are available it appears that the modulus of elasticity of firebrick at elevated temperatures lies between 500,000 and 2,000,000 psi. With 1,500,000 psi as an average and with 30,000 psi as the yield stress of steel, the following relation exists for elastic compression of brickwork:

$$\text{Unit compression} = \frac{30{,}000 \times \text{Cross-sectional area of binding}}{1{,}500{,}000 \times \text{Cross-sectional area of brickwork}}$$

$$= \frac{\text{Area of binding}}{50 \times \text{Area of brickwork}}$$

Since the overall unit expansion equals 4/1000, the elastic compression of the brickwork must take up 3/1000, if the binding stress is not to exceed the yield point. Then area binding = 50 × 3/1000 × area brick = 0.15 brick area.

It is economically impossible to provide binding with a cross-sectional area equal to almost one-sixth of the area of the refractory. This means that either expansion joints must be present, or the binding is stretched beyond the yield-point stress.

From the foregoing example it follows that the above-given equations furnish dimensions of binding that are greatly in excess of those used in practice. This means that, in all furnaces, the binding is stretched (as can frequently be observed) or that expansion allowance is present in the brickwork. If the binding has been stretched and the furnace is cooled, the bricks contract, either individually or in groups, and are not held together by the binding. Cracks open.

If, during the next heating period, each brick moved contrariwise, exactly the same distance as it did during the cooling period, the cracks would be closed and no binding would be needed. This ideal condition is, of course, never met in practice. In the corners, the bonding of the bricks interferes with orderly contraction. At each successive heating, some bricks project out from the walls at the corners, or else the corners lean outward. The binding must resist such movements, because they would lead eventually to collapse of the furnace.

Obviously, the relative movements of the bricks depend on fortuitous circumstances such as strength of bonding mortar, location of flues in walls, softening of the hot face by flames, and many other influences. The greatest force the binding (which had been stretched at the first heating up) could be called on to resist would equal the frictional resistance of one-half of one wall against its foundation. Even with a low working stress in the binding, an extremely small area of binding satisfies this condition. It is therefore advisable to introduce an empirical rule and to make the cross-sectional area of the longitudinal binding equal to ½ per cent of the brick area (sidewalls and roof) it supports. Thus it is apparent that considerable latitude is left to the furnace designer.

From the foregoing reasoning it is clear that the corners of furnaces must be protected and held rigidly. The steel angles that protect the corners are often extended below the furnace to serve as legs. Bending moments are minimized if the longitudinal or end binding is in line with the sidewalls. However, such an arrangement would, in many cases, locate the horizontal beams directly over the flues. Evidently, if the corner-to-corner horizontal beams are strong enough, the binding may be located in any convenient place. In the standard design of medium-sized furnaces, the cross beams under the hearth are welded to the buckstays and carry the plate that supports the hearth. Mention was made earlier that the bottom beams or tierods must not be buried in an unventilated hearth, if the roof is an arch.

If the beams and buckstays supporting an arch are rigid, any (or all) of three things happen: The arch rises, its hot face is compressed elastically, its hot face creeps (it is compressed plastically).

A suspended roof expands like a sidewall. No expansion joints can be provided. In most designs, the roof extends part way into the top of the walls. It is surrounded by a soft packing which is usually made of asbestos composition. In any event, a flat roof cannot exert a greater thrust against the binding than a sidewall of equivalent dimensions exerts.

So far as types of furnace-binding members are concerned, there is no end to the variety of forms, especially in home-built furnaces. Where tierods are used, two channels back to back frequently serve as buckstays. Where a single cross channel at the top and bottom replaces tierods, a heavier single channel is generally used for the buckstay. In many home-made furnaces, two old rails serve as buckstays, and angles are frequently used for corners. Buckstays near doors, and particularly between doors, are often exposed to heat and flame. The thin section of ordinary structural material does not carry heat away fast enough, and warping and burning result. For that reason, heavy rolled slabs are often used in the steel mills, and for high temperature furnaces the buckstays are frequently water-cooled.

The advent of the continuous strip mill reduced the price of sheet steel to such an extent that modern industrial furnaces can be encased in steel. The casing must be especially strong and rigid around portable furnaces, which have to withstand the bumping in a freight yard.

9. DAMPERS

Furnace pressure is controlled by an adjustable obstruction either in the flue which leads to the stack or else on top of the stack, if a stack is a part of the furnace. A damper on top of a flue appears in Fig. 288. The location of a flue damper can be seen in Fig. 289. In former years, dampers were generally located on top of stacks, because that location protected the dampers against destruction by hot gases or flames. However, rusting of the joint and changes of temperature of the operating rod make close automatic control of the furnace pressure impossible. For that reason, important furnaces are now equipped with flue dampers.

A flue damper is a plate which slides up and down across the main horizontal flue. The design and material of the damper must be adapted to the maximum temperature of the waste gases in which the damper hangs.

If the temperature of the waste gases does not exceed 1300 F, the ribbed damper (Fig. 290) serves very well. It is 3 in. thick over the ribs and is made of high grade nodular cast iron. For waste-gas temperatures up to 1600 F, castings of low alloy steel are preferred. Figure 291 illustrates, in detail, a widely used design. Warping is minimized by building a damper of horizontal sectional castings which are threaded on a valve stem of rectangular cross section. The bottom casting, 1, rests on taper, 2, of the valve stem. All other castings, 3, are alike. Each rests on the one below.

Air-cooled dampers have been tried. Radiation from flue gases and from adjoining brickwork is so great that air cooling does not protect the damper unless it is covered with insulation. The insulating material is held in place by studs.

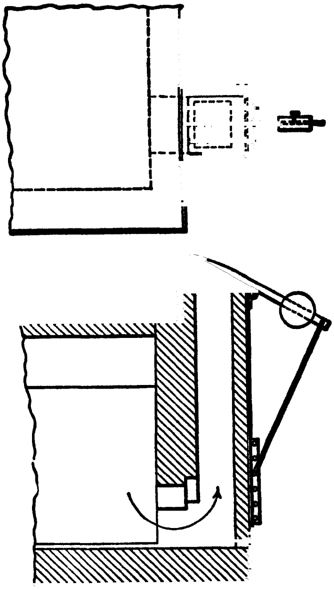


Fig. 288. Stack-type damper.

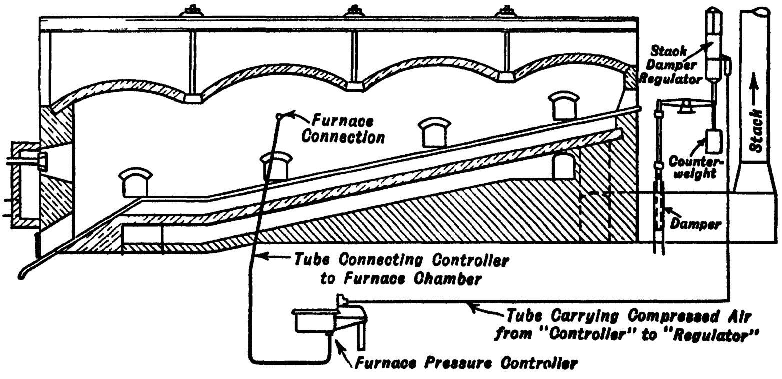

Stack Damper Regulator
Furnace Connection
Stack
Counter-weight
Damper
Tube Connecting Controller to Furnace Chamber
Tube Carrying Compressed Air from "Controller" to "Regulator"
Furnace Pressure Controller

Fig. 289. Automatic regulator for maintaining constant furnace pressure. (Courtesy of Hagan Corporation.)

For temperatures up to 2600 F, the design of Fig. 292 can be recommended. It was originated by the Blaw Knox Company of Pittsburgh. Bricks or tiles are shoved horizontally into the water-cooled frame immediately below the top crossbeam and are guided as well as held in place by ⅞-in. diameter rods which are welded to the vertical water-cooled members of the frame. The rods do not extend all the way to the top, so as to permit the insertion of tiles. The top tiles are locked in by short rods which are dropped into place through holes in the top crossbeam. These rods interlock with the crossbeam and with the tiles in the second row from the top. Since only a fraction of the damper surface is water-cooled, the consumption of cooling water is moderate. Incidentally, Fig. 292 shows a damper hood which reduces air infiltration to a low value.

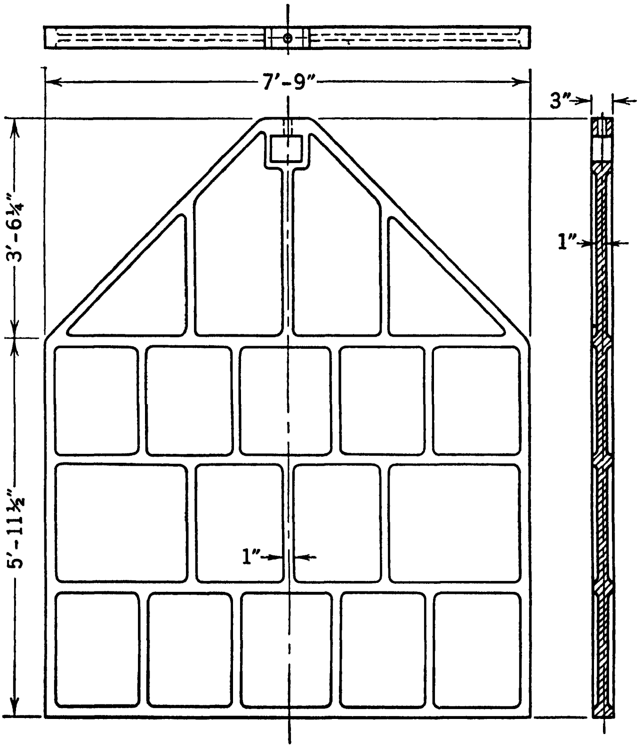


Fig. 290. Ribbed stack damper for low temperatures.

Temperatures in excess of 2600 F are not encountered in the waste gases from industrial furnaces in the sense of this book. Furnaces that discharge gases at higher temperatures are equipped with totally water-cooled dampers.

Heating furnaces of the batch type and heat-treating furnaces of all types are usually built without stacks because it is difficult accurately to control the atmosphere in such furnaces by attempting to control the considerable draft which is created by a stack. With the relatively small input of fuel involved, a sufficient tightness of the furnace and damper cannot be maintained to insure a positive furnace pressure required to prevent air infiltration. Waste gases are discharged from wall flues or from ports in the furnace roof to the atmosphere. Where automatic control of pressure is desired, these openings are often manifolded by means of a brick duct on top of the furnace,

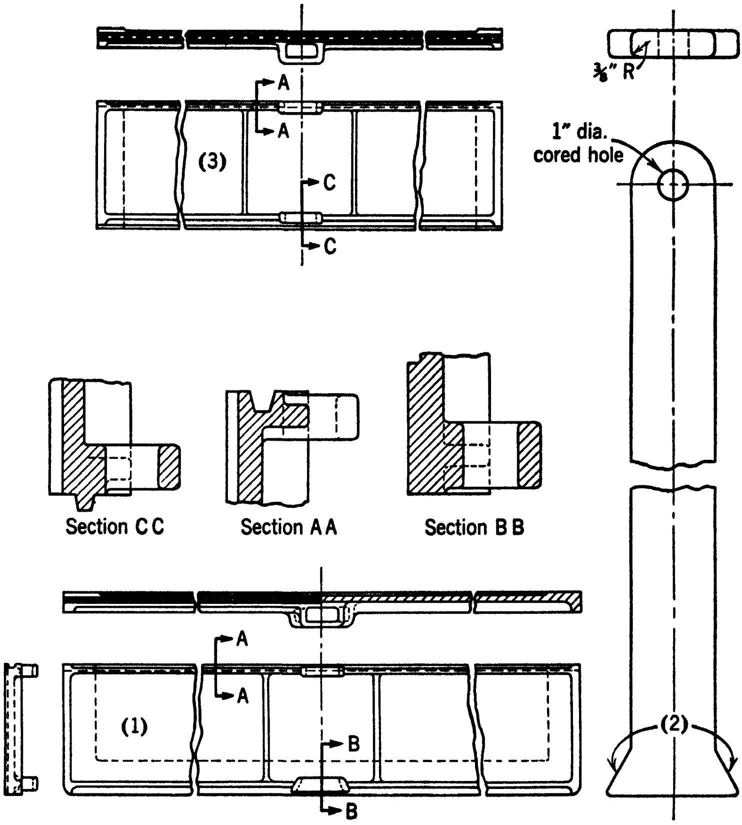


Fig. 291. Sectional (steel-castings) damper for medium temperatures.

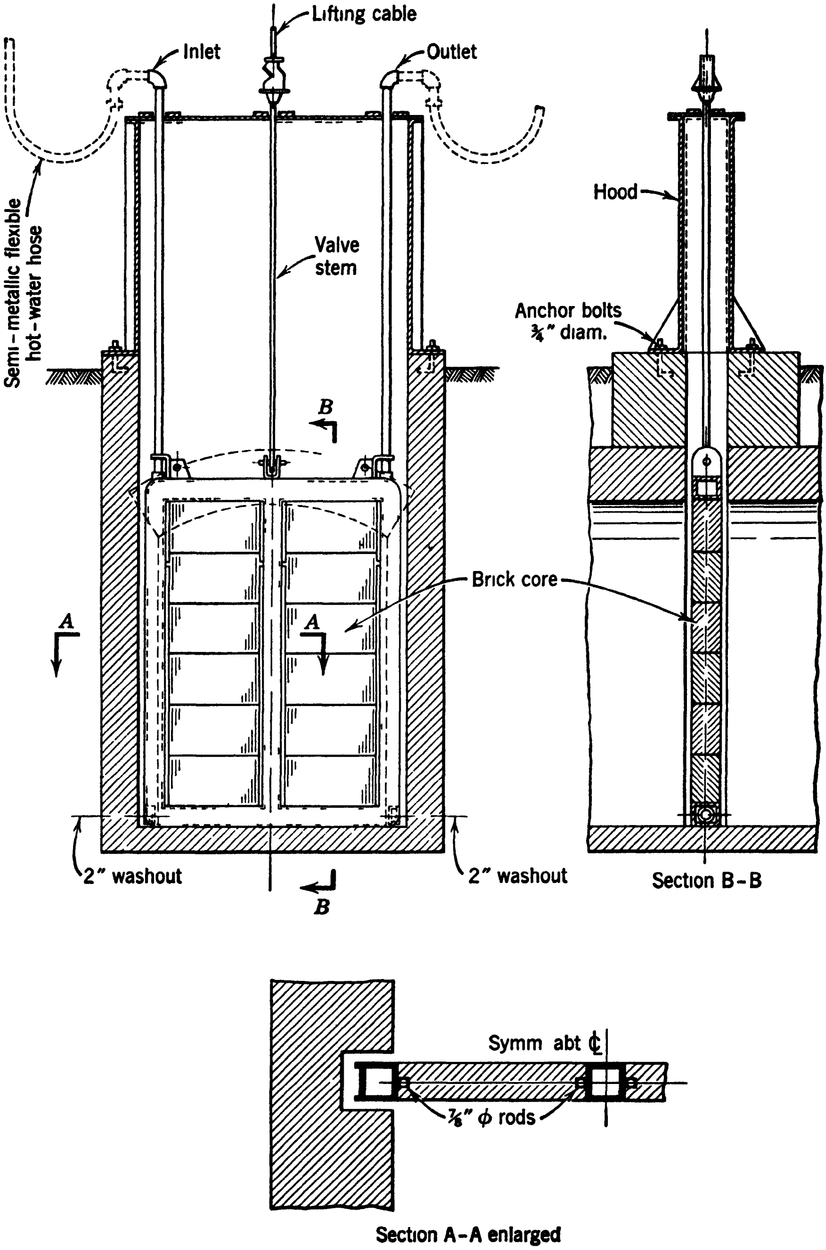
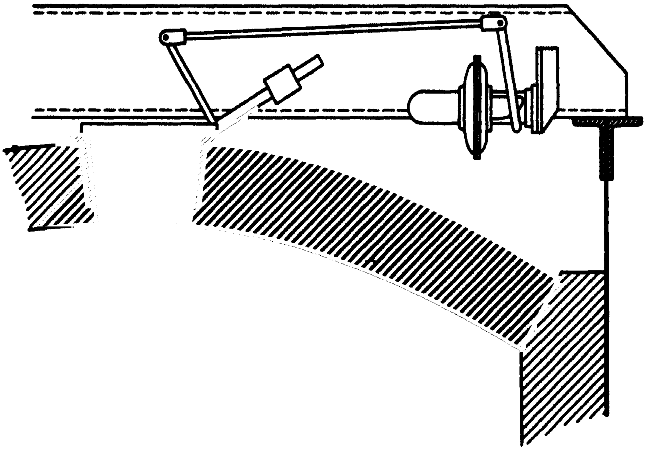


Fig. 292. Damper with water-cooled frame for high temperatures.

with one large opening provided with a mechanically operated damper. A typical air-diaphragm operated damper mechanism is illustrated in Fig. 293.

Since recuperators are an accessory, their strength and durability are discussed in the fourth edition of Volume II, and since tile recuperators and brick generators have many features in common, a discussion of the strength of regenerators is likewise transferred to Volume II.

Fig. 293. Air operated damper.



chapter **8**

Movement of Gases in Furnaces

1. PHYSICAL LAWS OF MOVEMENT OF GASES

The understanding of the important facts of this chapter will be very much facilitated by a review of the underlying physical facts concerning buoyancy, velocity head, friction between gases and solids, viscous drag, and induction. It is true that these laws may be found in almost any engineering handbook, but they are usually not in a form that is convenient for use by the furnace engineer.

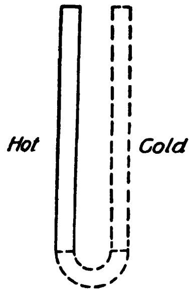


Fig. 294. Diagram illustrating cause of stack draft.

A column of hot air (see Fig. 294) weighs less than a column of cold air of equal height, which is shown dotted to form a U-tube. This dotted column corresponds, of course, to the outside atmosphere. The difference in weight produces a pressure difference which is known as draft and which is commonly expressed in inches of water. The draft is proportional to the height of the column and to the difference in density between the hot and the cold air. Expressed in pounds per square foot, the draft equals $(q_c - q_h)H$, where q_c and q_h are the densities of the cold and hot columns, in pounds per cubic foot, and H is the height of each column or stack in feet. Since 12 in. of water column exert a pressure of 62.3 psf (which numerically equals the weight of 1 cu ft of water), the draft, expressed in inches of water, equals

$$\frac{12}{62.3}(q_c - q_h)H$$

The densities of air and of other gases depend on their temperatures and pressures, thus: $q = p/(RT)$, where p is the absolute pressure in pounds per square foot and T is the absolute temperature. R is the gas constant which, for air, equals 53.3. The pressure of the atmosphere at an average elevation equals 14.3 psi abs. With this value for pressure, the density becomes $q = 38.6/T$, for air, and the draft, both columns being air, becomes

$$\begin{aligned}\text{Draft (in. of water)} &= \frac{12 \times 38.6}{62.3}\left(\frac{1}{T_c} - \frac{1}{T_h}\right)H \\ &= 7.43\left(\frac{1}{T_c} - \frac{1}{T_h}\right)H \qquad (30)\end{aligned}$$

If the hot column consists of stack gases, with a specific gravity of w_1 referred to air, the draft is $7.43(1/T_c - w_1/T_h)H$ in. of water.

If the equation for draft is to take care of variations of atmospheric pressure, either with altitude or with the weather, it becomes

$$\text{Draft (in. of water)} = 0.255B\left(\frac{1}{T_c} - \frac{w_1}{T_h}\right)H \qquad (31)$$

where B is the actual barometer reading in inches of mercury.

In Fig. 295, the solid line shows the draft, for a column 100 ft high and for cold air at 60 F, as a function of the temperature of the hot air. If the temperature in the hot column varies, the average temperature determines the draft. If the hot column contains products of combustion instead of air, a correction should be applied, because CO_2 is heavier than air and H_2O is lighter than air. The dotted line in Fig. 295 gives the draft, if the fuel is pure carbon and if perfect and complete combustion is attained. Most fuels are hydrocarbons, the combustion of which produces water vapor in addition to CO_2 The water vapor counteracts the effect of the greater density of the CO_2; accordingly the deviation from the solid line is very small. When the fuel contains much hydrogen or methane, the draft may be more than for air, as shown by the dot-dash curve for coke-oven gas. For any barometric pressure B, other than the standard of 29.92 in., the values read from the curve are simply multiplied by $B/29.92$. For stacks, the average temperature may be taken as the arithmetical mean between

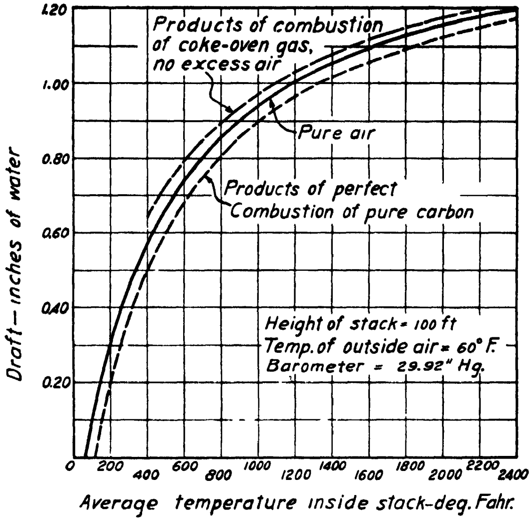


Fig. 295. Stack draft as a function of temperature.

top and bottom; for regenerators, if the absolute temperature increases at a uniform rate from T_1 at the bottom to T_2 at the top (degrees absolute), the effective average temperature (absolute) is

$$T_h = \frac{T_2 - T_1}{\ln(T_2/T_1)} \tag{32}$$

If the hot column be closed at the top and open at the bottom, the draft appears as an excess pressure in the hot column, that excess pressure being greatest at the top. If the hot column be closed at the bottom and open at the top, then atmospheric pressure will exist at the top, but pressure less than atmospheric (draft) will exist at the bottom of the column. If the temperature of the hot column be maintained constant with time, if the hot column be open at both ends, and a resistance be offered to flow, as, for instance, by means of the orifice 1 in Fig. 296, then the draft establishes a flow through the column in such a manner that the draft is balanced by the resistance to flow, which is the sum of all velocity heads plus friction heads.

If the orifice 1 be quite small in diameter compared to the column, all the draft is converted into velocity head in the orifice. Draft, velocity head, and density of gas flowing through the orifice are correlated by the equation*

$$\text{Draft loss (in. of water)} = \frac{(\text{Velocity})^2}{335 \times \text{Specific volume}} \tag{33}$$

in which the velocity is expressed in feet per second and the specific volume in cubic feet per pound. (This equation refers to the actually obtained maximum velocity in the contracted vein.) For air, the equation becomes†

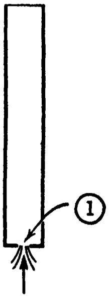


Fig. 296. Orifice at base of stack.

$$\text{Draft loss} = \frac{0.118c^2}{(T + 460)} \tag{34}$$

This relation is graphically expressed by the charts (Figs. 297 and 298). For any gas other than air, the pressure drop is proportional to the specific gravity.

If the diameter of orifice 1 be such that it cannot be called very small compared to the diameter of the column, friction in the column becomes a factor. The pressure loss by friction in a round pipe is commonly expressed by the formula:

$$p_1 - p_2 = f \cdot \frac{L}{D} \cdot q \cdot c^2 \tag{35}$$

where $p_1 - p_2$ = pressure loss, inches of water; f = a friction coefficient; L = length of pipe, feet; D = diameter of pipe, feet; c = average velocity of flow, feet per second; q = density, pounds per cubic foot = $1/v$ = 1/specific volume.

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* $335 = \frac{62.3 \times 2g}{12}$

† $0.118 = \frac{14.3 \times 144 \times 12}{53 \times 2g \times 62.3}$

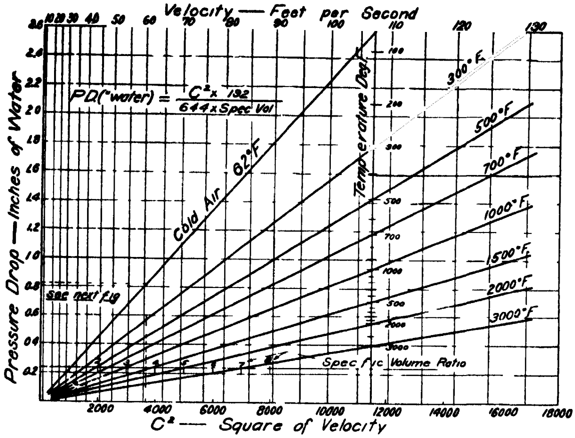

Fig. 297. Relationship of velocity, specific volume, and draft loss for high velocities.

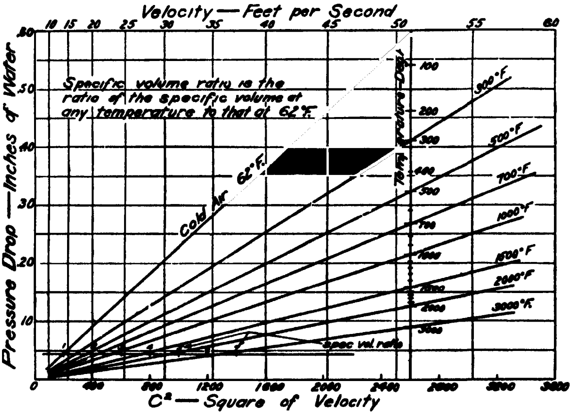

Fig. 298. Relationship of velocity, specific volume, and draft loss for low velocities.

For average conditions of turbulent flow, equation 35 is commonly written in this form

$$p_1 - p_2 = \frac{Lc^2q}{13{,}300D} \text{ in. of water} \qquad (36)$$

if D is expressed in feet, or

$$p_1 - p_2 = \frac{Lc^2q}{1110D} \tag{37}$$

if D is expressed in inches.

If the ducts are extremely rough, the pressure drop may be up to 45 per cent greater than the value furnished by equation 36. And, if the ducts are very smooth, the pressure loss may be as much as 30 per cent lower than the value obtained from equation 36. Compare the right-hand section of Fig. 300.

In furnace work, round ducts are seldom encountered, except in stacks. The pressure loss in a rectangular duct is given by the equation

$$p_1 = p_2 = fLqc^2\frac{1}{2}\left(\frac{1}{D} + \frac{1}{b}\right) \text{ in. of water} \tag{38}$$

In this equation, D and b are the sides of a rectangle.

Although equation 35, making use of a constant coefficient of friction, has been widely applied, such use leads to errors of considerable magnitude under some conditions. If the formula is to hold over a wide range, the coefficient f cannot be taken as constant but must vary

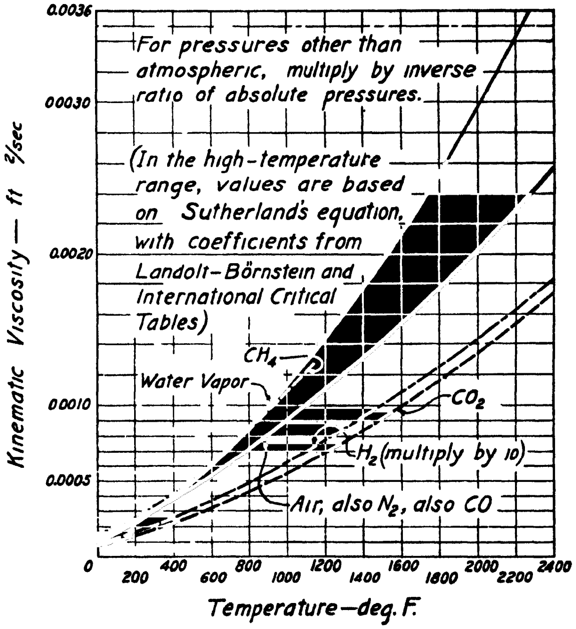

Fig. 299. Kinematic viscosity of air and other gases, at atmospheric pressure.

with the expression cD/u, where c and D have the same meaning and units as previously and where u is the kinematic viscosity of the gas in square feet per second. Tables of viscosity are not readily available, and those that have been published are usually given without the units. To remedy this condition, Fig. 299 is offered; it gives the kinematic viscosity of air and other gases at atmospheric pressure as a function of the temperature. In a mixture of gases, the viscosity does not vary in direct proportion to the percentages of the components; if these have nearly the same density, the viscosity of the mixture may be greater than that of any of the components. For flue gases, the viscosity is very nearly the same as for air. For fuel-gas mixtures containing much hydrogen, the kinematic viscosity does not increase in proportion; it may be taken roughly as equal to (viscosity of heavy gas that is present in greatest percentage) × (ratio of specific volume of mixture to specific volume of heavy gas).

Figure 300 gives the variation of the friction coefficient (in equation

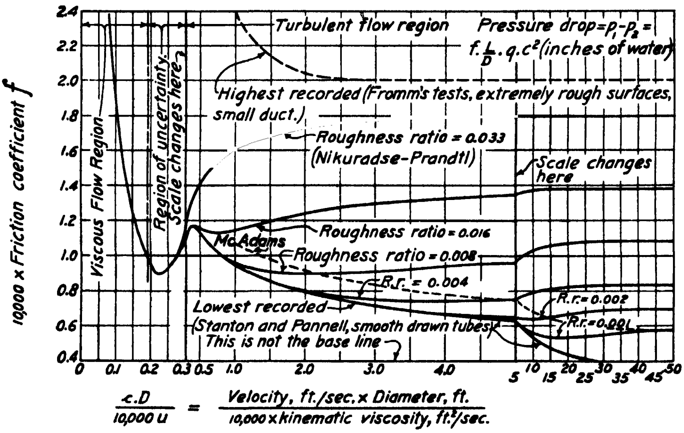

Fig. 300. Friction coefficients for flow of fluids in pipes or ducts.

$p_1 - p_2$ = pressure drop, inches of water.
L = pipe length, feet.
D = diameter of circular pipe, feet, or hydraulic diameter of any duct, feet.
c = velocity, feet per second.
q = density of fluid, pounds per cubic foot.
u = kinematic viscosity, square feet per second.
f = frictional resistance of coefficient, from curves.

$$\text{Roughness ratio} = \frac{\text{average height of projections in walls of ducts}}{\text{diameter of duct}}.$$

Broken-line curve marked "McAdams" applies to commercial iron or steel pipes.

35) as a function of the term cD/u, and for several different degrees of roughness of the pipe or duct. Carefully conducted tests of friction losses of hot gases in brick-lined ducts are not available, but from very convincing reasoning in the theory of the flow of fluids it follows that the coefficients must lie among the curves of Fig. 300. The "roughness ratio" marked on the curves means the ratio of the average height of the waves or projections in the walls to the diameter or width of the pipe or duct. Fairly smooth walls corresponding to brickwork that has been laid evenly and has been brush-coated with a cement would have a roughness ratio of about 0.004. For medium-rough walls, corresponding to brickwork that has been laid evenly without such coating, the ratio may be taken as 0.010, whereas, for brickwork that has been unevenly laid, or brickwork that is coated with slag waves, the roughness ratio may be as high as 0.04. It must also be noted that a given amount of unevenness of surface in a large duct corresponds to a smoother pipe than it does in a small duct, because roughness is measured by the ratio of depth of projections to diameter of duct.

For values of cD/u less than 50,000, the coefficient of friction varies considerably. When cD/u is less than 2000 (abstract number) the stream lines become parallel, and no turbulence exists. The pressure drop in this region is proportional to the first power of the velocity. In order to make equation 35 apply to this region also, the coefficient of friction is made to rise rapidly with diminishing cD/u.

EXAMPLE. A brick-lined stack, 70 ft high, and 27 in. inside diameter, carries off flue gases having 1400 F temperature at the bottom of the stack and 1200 F at the top of the stack. The stack damper is almost closed at the top, being raised only 1 in. from its seat. If there is no resistance between furnace and stack, how much flue gas passes up the stack in 1 min?

The density of the mixture of flue gases will be taken as equal to that of air.

Average temperature in stack = 1300 F

$$\text{Area of cross section} = \frac{\pi}{4}\left(\frac{27}{12}\right)^2 = 3.98 \text{ sq ft}$$

$$\text{Area through damper } \frac{\pi \times 27 \text{ in.} \times 1 \text{ in.}}{144} = 0.59 \text{ sq ft}$$

Taking 0.62 as the coefficient of discharge,

$$\text{Ratio } \frac{\text{velocity through damper, } c_1}{\text{velocity in stack, } c_2} = \frac{3.98}{0.59 \times 0.62} \times \left(\frac{1200 + 460}{1300 + 460}\right) = 10.2$$

(*a*) Draft required for producing velocity through damper

$$\frac{0.118c_1^2}{1200 + 450} = 0.000071c_1^2$$

(*b*) Draft required for overcoming friction in stack

$$= f\frac{L}{D}qc_2^2$$

or,

$$\text{since } q = \frac{39.7}{T + 460},$$

$$= f \times \frac{70}{2.25} \times \frac{39.7}{1300 + 460} \times c_2^2 = 0.701fc_2^2$$

(*c*) Draft produced by buoyancy (gases of same density as air)

$$= 0.255 \times 29.92 \text{ in. mercury} \times 70 \text{ ft} \times \left(\frac{1}{60 + 460} - \frac{1}{1300 + 460}\right)$$

$$= 0.723 \text{ in. of water}$$

Equating c to a plus b gives

$$0.723 = 0.000071 \times (10.2c_2)^2 + 0.701f \cdot (c_2)^2$$

Taking $c_1 = 100$ as a trial value, and $c_2 = \frac{100}{10.2} = 9.8$

$$u \text{ (from Fig. 299)} = 0.0012 \quad \text{and} \quad \frac{c_2D}{u} = \frac{9.8 \times 27}{0.0012 \times 12} = 18{,}400$$

From Fig. 300, taking values intermediate between the curves for roughness ratios of 0.008 and 0.016,

$$f = 1.05 \times 10^{-4}$$

$$0.723 = 0.000071 \times (100)^2 + \frac{0.701 \times 1.05}{10{,}000} \times (9.8)^2$$

$$0.723 = 0.710 + 0.0071 = 0.717$$

The trial velocity is thus shown to be practically correct.

$$\text{Quantity} = 3.98 \text{ sq ft} \times 9.8 \text{ fps} \times 60 = 2340 \text{ cfm at } 1300 \text{ F}$$

$$\text{Weight} = 2340 \times \frac{39.7}{1300 + 460} = 52.8 \text{ lb per min}$$

Caution is necessary, if equation 35 is to be applied to the flow of gases through a furnace interior. In the flow of gases through a furnace of the type illustrated in Fig. 301, the wavy roof causes so much turbulence that the coefficient of friction is very great. And the billets lying on the hearth are more or less crooked, which fact also contributes to increasing the coefficient of friction. If the gas flow fills the cross section of such a furnace, the pressure drop may be twice to five times as high as any value found from equation 35, depending on conditions.

In many other furnaces, the gas flow does not fill the cross section of the heating space but only the upper portion. The stock is almost wholly heated by radiation. In such a case, friction between gases and solid walls is limited to the roof and the upper sections of the sidewalls. Calculation then follows the same rules that are used for flow of water in open channels. However, the problem is more complicated, because density and velocity change continually.

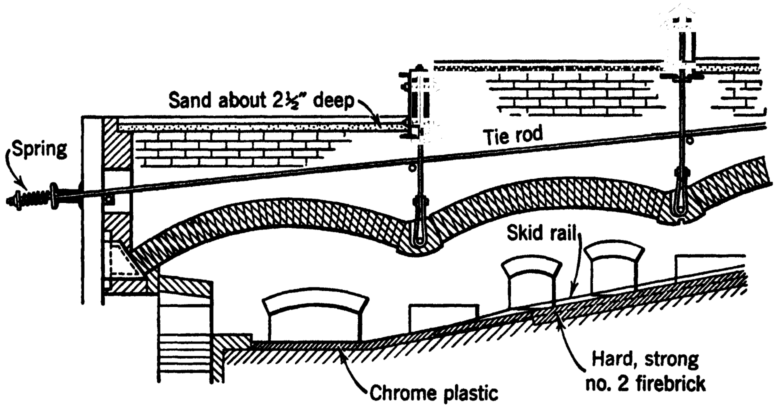

Fig. 301. Typical roof construction for a wide continuous-heating furnace, showing a spring-loaded tie rod.

For these and other special cases, some experimenters have derived special equations. As a rule, density, velocity, and linear dimensions enter their equations with fractional exponents. Such equations are useful for a very narrow range of conditions only, and for conditions that are almost identical with those under which the tests were conducted.

Overly great accuracy in determining the frictional pressure drop in straight ducts is out of place in furnace work. Neither the rate of flow nor the gas temperature nor the relative roughness is accurately known. And furthermore, the sum of the pressure drops at bends, enlargements, and contractions of section, and through valves almost always exceeds by far the drop due to friction in the straight sections of ducts. These other losses are expressed either as fractions (or multiples) of the velocity head or else as pressure drop through an additional length of duct. Expressing the pressure drop by means of the velocity head is very convenient, because, by means of the charts (Figs. 297 and 298), the velocity head is quickly converted into pressure drop or draft loss, expressed in inches of water.

The method of expressing pressure drops through bends etc. as equivalent to an additional length of duct is common in those branches of engineering that employ circular sections (that is to say, pipes) for conducting fluids. In consequence, tests on equivalent lengths are limited to circular sections. Items 1 and 2 in Fig. 302 and also Figs. 303 and 304 indicate that the pressure drops for round and square sections differ. For that reason, equivalent lengths of pipe are not given here. They are found in any engineering handbook.

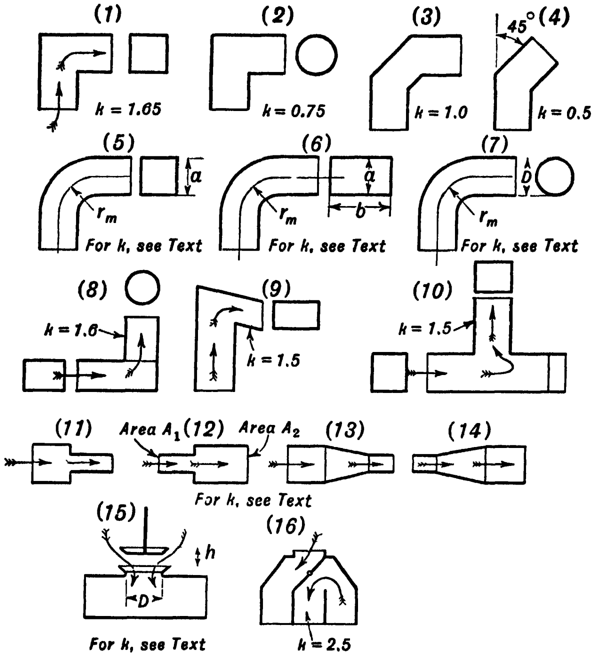

Fig. 302. Loss of pressure or draft at various changes of section in ducts. k = pressure drop as a multiple of velocity head

The factors applying to various shapes of duct and forms of section change are shown in Fig. 302.* For right-angle bends in rectangular ducts, the loss is less than in square ducts if the short axis of the cross section is in the plane of turning (Fig. 302, 6) and greater than for the square if the long axis of the rectangle is in the plane of turning; the factors of Fig. 302, 1 are therefore to be multiplied by correction factors as in Fig. 305, for rectangular ducts. If the bends are rounded instead of sharp-cornered—i.e., elbows, as in Fig. 302, 5 and 6—the loss factors vary with the ratio of the mean radius of elbow to the width or diameter of the duct, as shown in Figs. 303 and 304.

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* Wirt, *General Electric Review*, June 1927; Nippert, *Forschungarbeiten Vereines deutscher Ingenieure*, 1929, Heft 320; Heil, *Archiv für das Eisenhüttenwesen*, Volume 12, 1928, page 729; authors' tests.

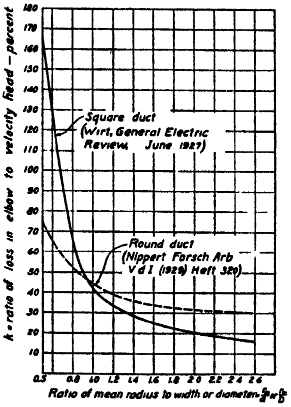

Fig. 303. Loss of pressure or draft at 90-degree elbows in square or round ducts.

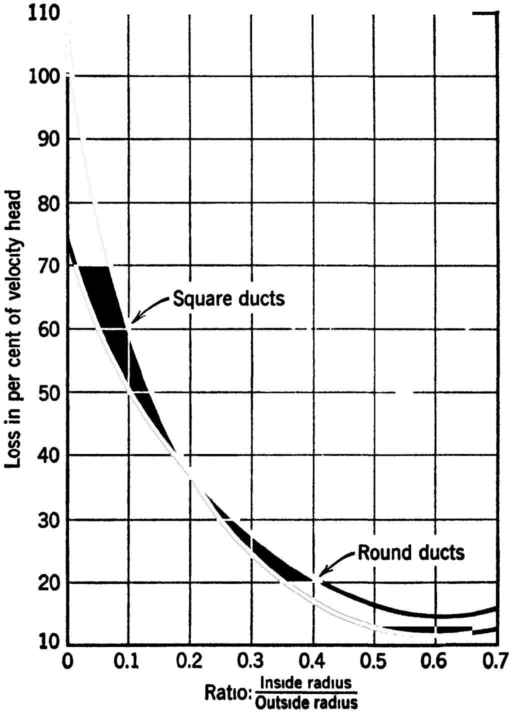

Fig. 304. Loss of pressure or draft at 90-degree elbows in square or round ducts.

Figure 304 is based on tests that were made in the laboratories of the Buffalo Forge Company and were published in "Fan Engineering," fifth edition. The abscissae of Figs. 303 and 304 can be converted into each other by the relation

$$\frac{\text{Mean radius}}{\text{Width of duct}} = \frac{r + R}{2(R - r)} = \frac{r/R + 1}{2(1 - r/R)}$$

The length of straight duct beyond the 90-degree elbow and the location of the pressure taps affect the test results.

On account of the added expense in bricklaying, masonry elbows with curved walls are comparatively rare.

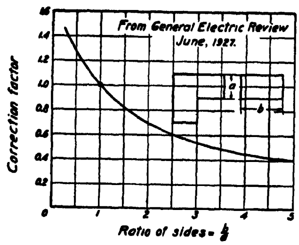

Fig. 305. Relation of loss at bends in rectangular ducts to loss at bends in square ducts.

In a contraction of section such as 11 of Fig. 302, the loss depends on the area ratio A_2/A_1 and on the sharpness (or rounding) of the edges at the entrance. For sharp edges, the pressure drop (ft head) is commonly given to equal $kc_2^2/2g$, where k varies with the area ratio, in the manner indicated by the following tabulation:

A_2/A_1	0.9	0.8	0.7	0.6	0.5
k	0.075	0.15	0.22	0.26	0.30

If the entrance corners are rounded, for instance by the use of door-jamb bricks, the pressure drop is smaller. If the contraction occurs in a converging passage as at 13, the pressure drop varies with the included angle, but for usual proportions it is 1.4 times the difference between the final and initial (exit and entrance) velocity heads. Where the gas flow enters an enlarged section as at 12, the loss of head (ft) is commonly computed from the expression $\frac{(c_1 - c_2)^2}{2g}$. No rational explanation exists for this formula; however, it agrees satisfactorily with test results. If the sudden enlargement is followed by a sufficiently long straight duct of constant cross section, roughly 40 per cent of the pressure drop is recovered. In a diverging passage, such as at 14, there occur both a recovery of pressure (Bernoulli's theorem) and a drop of pressure (caused by surface friction). For included angles of 4 or 5 degrees, and for very smooth walls, the pressure rise may reach 80 per cent of the difference of the velocity heads. Fan-tail arches in regenerative furnaces are an example of converging and diverging flow.

The drop of pressure in saucer valves, such as 15, is shown in Fig. 306.

The draft loss through a series of resistances, such as a port, flues, and stack with several bends, is found as the sum of the individual losses. An example is given under "Flues and Stacks."

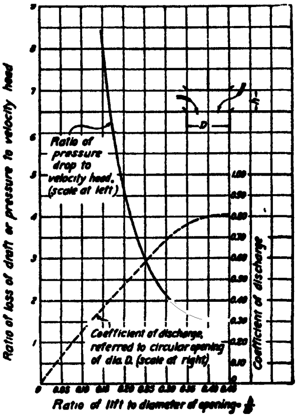

Fig. 306. Loss of pressure and coefficient of discharge for saucer valves.

If a current of air, or fluid, such as is caused by the products of combustion of gas or oil burners, passes through space, it gathers onto itself particles of the surrounding fluid, imparting velocity to them by viscous drag. The main stream slows down in such a manner that the product of moving mass and velocity remains approximately constant. The total angle of the cone that envelops the moving mass varies with the initial velocity of the jet. In cold air, it is about 16 degrees for slow jets (10 fps), increasing gradually to about 25 degrees for jets traveling at more than 1000 fps. When a jet of cold air induces hot air or combustion gases, the jet expands at greater angles than in cold air. By these statements, the movement of the jet is sufficiently well defined for practical purposes, but the following additional statements will also be of interest: The velocity at the edge of the jet equals zero, and the velocity at the center is approximately equal to twice the average velocity.

In the application of these rules to furnace jets, care must be taken to apply them only to currents in which combustion has been completed, because changes of specific volume due to combustion affect the result considerably.

The theory of jet induction is further discussed on pages 415–416. Applications are given in the second volume.

If a gas or air current passes along a furnace wall or along the surfaces of the stock, it is retarded by both viscosity and turbulence. The retardation due to turbulence grows with the roughness of the surface of the wall. No tests of the effects of furnace walls on circulation are available, as far as the authors know. In dealing with this problem, engineers must, at least for the present, rely on experience and judgment rather than on mathematical derivations. Judicious application of the laws of the flow of water in a wide, shallow channel may be of assistance.

For completeness, the gain in pressure by deceleration or slowing down of the flow should be mentioned. In passing through a regenerator, or in moving upward through a tall stack, the gases decrease in temperature and therefore move more slowly. This is equivalent to a gradual enlargement of the cross section, as in a Venturi tube, for flow at constant temperature; consequently, some of the kinetic energy is transformed into pressure energy. The maximum amount of pressure regainable in a frictionless tube corresponds to the difference of the velocity heads (Fig. 298) between the initial and final velocities. The recovery of pressure due to this cause is so very small, however, in comparison to the frictional pressure drop, that in the majority of practical cases it is neglected.

2. FURNACE PRESSURE; SIZE AND LOCATION OF PORTS

In connection with the pressure conditions in furnaces, two important rules must be observed. They are as follows:

1. In the heating of metals, the pressure in the heating chamber must be atmospheric, or only very slightly in excess of atmospheric, at all rates of heating.
2. The lower the temperature to which the material is to be heated, the greater is the necessity for thorough circulation of the gases in the heating chamber, especially if piled or coiled material is to be heated rapidly and uniformly.

The justification of these rules, their explanation, and their application are an important part of the present chapter.

The reason why pressure in the heating chamber must, for correct operation, be practically atmospheric is obvious. If it were much greater than atmospheric, flame or hot gases would burst out of all openings, and, if it were smaller, air would be drawn in through the doors, and the material would be oxidized and scaled. It should, however, be mentioned that air in the heating chamber is not injurious, if ceramic ware is being heated.

In a tall furnace, it is impossible to have atmospheric pressure at all levels, because the furnace forms a chimney or stack (column of hot gases). Evidently, there must be a difference between atmosphere and furnace pressure at some level of the furnace; and the question is, "How should that pressure difference be distributed?" For the purpose of preventing the entrance of air, furnace pressure is almost invariably in excess of atmospheric pressure from the hearth up, the excess at the hearth being as little as 1/200 in. of water pressure, while the excess pressure at the crown of the arch is much greater, depending on height and temperature of the furnace. Furnace operators test the furnace pressure that prevails near the hearth by watching for a "stinger" of flame or hot gas issuing from an observation hole in a furnace door. Fine dust dropped in front of the door opening intensifies the indication.

The desirable condition of carrying a small excess pressure at the level of the hearth is easily maintained if the products of combustion leave near the hearth. This arrangement has the advantage that relatively cool products of combustion lying on the charge are swept out through the vent. The flow of gases in a furnace equipped with burners near the roof and with vents near the hearth is illustrated by Fig. 307. With this arrangement, the flame (or heat) licks the hearth. The U-shaped travel of the products of combustion gives time for the products to give up their heat and to leave with but little excess temperature.

If the products of combustion leave through the roof, and if the burners are located above the heating stock, the products of combustion leave the furnace while still burning and without having time to give up their heat to the charge. If the vent-opening is large enough for high fire, it is too large for average firing. A slight vacuum is generated in the furnace, and cold air leaks in; this condition would be very severe in tall furnaces that carry high temperatures. Vents in the roof are avoided in such furnaces.

In certain underfired furnaces carrying temperatures between 1300 and 1700 F, vents in the roof are correct engineering. From earlier chapters it is known that, in the 1300 to 1700 F temperature range, radiation is weak and must be supplemented by convection for the purpose of increasing heat transfer and temperature uniformity. Figure

308, which is similar to Fig. 7, indicates how this goal is reached in spite of a roof-vent. The jet of flame coming from a burner inspirates cooler products of combustion and circulates a tempered flame over the charge. With either design (Figs. 307 or 308) correct furnace pressure is maintained either by an automatic damper, Fig. 293, or by a manually operated damper, Fig. 288, or, in small furnaces, by a brick that adjustably reduces the discharge opening of the vent or flue.

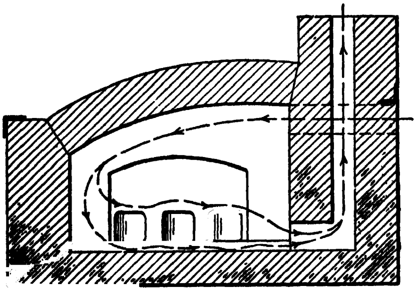

Fig. 307. Location of vent near hearth.

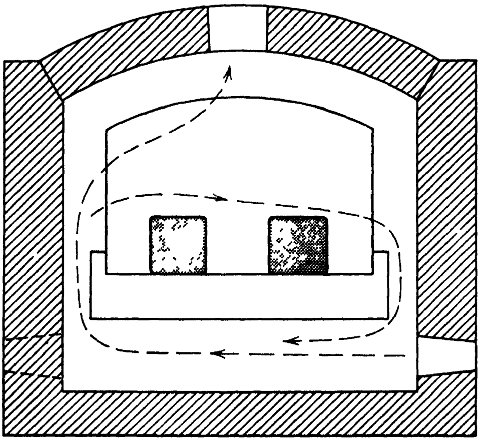

Fig. 308. Circulation in underfired furnace.

Method and apparatus for automatic control of furnace pressure are discussed in the fourth edition of Volume II.

Every flue is a stack that draws cold air into the furnace whenever a furnace door is opened, unless a damper enters into action simultaneously. The entering air is made harmless if the vent is located close to the door, because in that case the air is short-circuited through the vent into the flue.

For the sake of temperature uniformity, long furnaces of the batch type have a plurality of vents and flues. The artifice of locating a vent near the door cannot be used in such a furnace. Automatic control of furnace pressure, with a damper on each flue, is complicated and expensive. For that reason, ingenious furnace engineers have devised means for semi-automatic control. Two of these means are worthy of mention: One consists in providing a small, adjustable opening at the foot of each flue, outside the furnace. The other means consists in constructing the flues of very thin firebricks. The purpose of each artifice is the same, namely, to produce less draft at low rates of firing than is produced when the furnace is fired to capacity.

For the purpose of making vents and flues slightly oversized, the dimensions that are needed for maximum rate of firing must be determined. The required dimensions are a function of volume and temperature of products of combustion flowing in unit time and of the permissible drop of pressure. For a given furnace, hourly weight (and volume) of products of combustion is found from Chapters 3 (furnace capacity) and 5 (fuel economy).

In these calculations, the following fact is helpful: The release of a given number of heat units results in an almost constant weight of products of combustion, regardless of the type of fuel (with the exception of blast-furnace gas). In consequence, the volume of flue gases passing through vents and flues in unit time is known with sufficient accuracy, if the Btu's released in unit time and the temperature of the flue gases are known. Roughly, the release of 100 Btu produces 1 cu ft of products of combustion, referred to atmospheric pressure and room temperature. The rest of the calculation coincides with that given in the next paragraphs, which are based on the material heated on the hearth in unit time.

Very few furnaces heat more than 100 lb of steel per (sq ft, hr), 60 lb per (sq ft, hr) being considered a good rate, whereas underfired furnaces seldom heat more than 35 lb.* These figures, as well as the following derivations, apply only to furnaces of the "in-and-out," or batch, type.

The heat requirement of the steel, per hour, is easily found from this rate and from the heat content per pound (see Fig. 27). It approximately equals 160 Btu per lb at 1200 F, 250 Btu per lb at 1600 F, and 360 Btu per lb at 2200 F. On the basis of the very high rate of 100 lb per sq ft and hr, these figures correspond to 16,000, 25,000, and 36,000 Btu per (sq ft, hr). Radiation losses and convection losses usually equal 15 per cent of these values, except in underfired furnaces, in which the losses reach 20 per cent, or even a little more. If the ratio of the heat given off in the furnace by the gases to the total heat value of the fuel is called the gas efficiency, then the latter equals about 44 per cent for 2200 F furnaces, 59 per cent for 1600 F furnaces, and 69 per cent for 1200 F furnaces. Hence, the heat consumption in the fuel will be $1.15 \times 16{,}000/0.69 = 26{,}700$ Btu per (sq ft of hearth, hr), for a 1200 F furnace, with corresponding values of 48,700 for a 1600 F furnace, and of 94,000 for a 2200 F furnace. To produce 10,000 Btu by combustion requires roughly 95 cu ft of air (measured cold). The volume of the flue gases (referred to the same temperature as that of the combustion air) is very little more than that of the air, except in the case of blast-furnace gas and producer gas, so that we may figure with 105 cu ft of flue gas (cold measure) per 10,000 Btu. The ports then have to pass $\frac{26{,}700}{10{,}000} \times 105 \times \left(\frac{1200 + 460}{520}\right) = 900$ cu ft of hot flue gases per sq ft of hearth and hr for a 1200 F furnace. In the same manner, we find 2040 cu ft per hr for 1600 F furnace, and 5000 cu ft per hr for a 2200 F furnace.

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* It should again be noted that the method of rating the capacity of a furnace in pounds of steel per square foot of hearth and hour is very crude and that it is useful for quick approximations only. The method of loading the furnace, shape of pieces being heated, and uniformity of temperature in stock affect the rate of heating, as was explained in Chapter 2.

From these figures the port and flue sizes are easily calculated, if the velocity at which the gases may travel through them is known. That velocity is determined by the resistance that ports, flues, and stacks offer to the flow. A port which is nothing more than a hole in the roof or sidewall offers resistance due to the formation of velocity head only. A very short flue can be treated as a port.

If the pressure in the furnace at the port is not to exceed 0.01 in. of water, and if no stack is available to produce a draft, the velocity is limited (see Fig. 298) to 11.8 fps at 1200 F, to 13¼ fps at 1600 F, and to 15 fps at 2200 F. These figures refer to the mean velocity at the section of highest average velocity (contracted vein, Fig. 309), and not to the so-called average velocity. The latter is about 62 per cent of the highest velocity, on account of the inertia of the air or gas particles, as indicated by the arrows 1 in Fig. 309, unless the entrance to the port be well rounded. If the gases, instead of issuing through a thin-wall opening into the air, pass into a flue (as is usually the case) then the so-called average velocity lies in the range between 62 and 85 per cent of the average velocity at that contracted vein, according to local conditions.

The figures of this paragraph prove convincingly that "holes in the wall" must be quite large if they are to take care of the products of combustion. To make them smaller, and also to protect furnace attendants from heat issuing at hearth level from the furnace, short vertical flues are connected to the ports. By bearing in mind the facts given in the beginning of this chapter, it is possible to calculate the dimensions of ports and flues in such a manner that draft plus furnace pressure will just be balanced by the resistance of ports and flues. A knowledge of these facts will also make it easy to understand the following calculation for determining those velocities through ports and flues which are to be used under different conditions.

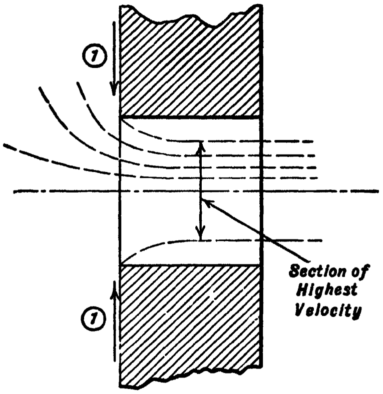

Fig. 309. Contracted vein in sharp-edged ports.

The static, or "no-flow," draft produced by a stack H ft high, with an average temperature of T F of the stack gases, when the temperature of the outside air is 60 F, barometer 29.92 in., may be expressed by the formula $0.01467H\left(1 - \frac{520 \times q_g}{(T + 460)q_a}\right)$ in. of water, or it may be taken from the chart (Fig. 295). In the above expression, q_g/q_a is the ratio of density of flue gas to density of air at the same temperature and pressure.

The draft loss caused by friction in a stack and flue of constant cross section is $\frac{q_g}{q_a} \times \frac{119.1 \times f \times B \times L \times c^2}{A(T + 460)}$ in. of water, if A = free cross-sectional area of the duct in square inches; B is the perimeter, or the circumference of the passage, in inches; L is the length of flue plus stack in feet; c is the velocity of the gases in feet per second; and f is a frictional resistance coefficient of which the values are shown in Fig. 300.

The velocity head (taking care of the coefficient of contraction) is $\frac{q_0}{q_a} \times \frac{0.184c^2}{(T + 460)}$ in. of water. The added resistance caused by a sharp right-angle bend in a square duct is $\frac{q_0}{q_a} \times \frac{0.195c^2}{(T + 460)}$ in. of water (see also Figs. 303 and 304).

In preparing Table 22, the length L was taken as 120 per cent of the height H. The pressure in the furnace was taken as 0.02 in. of water. The ratio q_0/q_a was taken as unity. Balancing the available draft against the sum of all losses or resistances gives the equation

TABLE 22. Velocities in Flues

Temperature in Stack, T F	1200			1600			2200		
H = height, ft	3	8	20	3	8	20	3	8	20
Size of Flue, in.	Permissible Velocity in Flue and Stack, fps								
4½ × 4½	13.9	18.0	22.8	15.8	20.7	26.4	18.3	24.1	30.7
4½ × 9	14.1	18.6	24.5	16.1	21.5	28.3	18.7	25.2	33.0
9 × 9	14.4	19.4	26.4	16.4	22.4	30.5	19.1	26.2	35.8
18 × 18	14.6	20.2	28.6	16.7	23.3	33.0	19.4	27.3	38.9

$$0.01467H\left(1 - \frac{520}{T + 460}\right) + 0.02 = \frac{c^2}{(T + 460)}\left(0.184 + 0.195 + \frac{119.1fB \times 1.2H}{A}\right)$$

or

$$0.01467H\left(1 - \frac{520}{T + 460}\right) + 0.02 = \frac{c^2}{(T + 460)}\left(0.379 + 142.9 \times \frac{fBH}{A}\right) \qquad (39)$$

To solve this equation directly for c, it would be necessary to assume a constant value for f. However, f is not quite constant, which fact makes the calculation more tedious. Table 22 was computed on the basis of the coefficients shown in Fig. 300.

The tabulation is very instructive. It shows that, for a very small furnace (low flue, small cross section) and for low temperatures, the velocity through the flues and ports must be low (14 fps), if excessive furnace pressure is to be avoided. It also shows that in large furnaces with high temperature, velocities up to 40 fps are permissible.

In view of these results it appears almost hopeless to formulate a simple rule for port sizes that is applicable to all furnaces. However, for quick estimates, and for "sell-across-the-counter-for-any-purpose" furnaces, the following reasoning will be helpful. From the tabulation, it follows that velocities of 19, 23, and 27 fps are good averages for furnaces of 1200 F, 1600 F, and 2200 F, respectively. On that basis, the following results are obtained.

Furnace temperature, F	1200	1600	2200
Port area (also flue area), sq in. per sq ft of hearth (heating steel)	$\frac{900^* \times 114}{19 \times 3600} = 2.0$	$\frac{2040^* \times 144}{23 \times 3600} = 3.5$	$\frac{5000^* \times 144}{27 \times 3600} = 7.4$

* The numbers 900, 2040, and 5000 represent cubic feet of hot gases per (hr, sq ft of hearth area), as calculated on page 384.

These figures are necessarily approximate. In practice, deviations are found, both up and down. They are partly caused by the practice of making ports and flues equal to multiples of brick sizes. Furthermore, the port areas were based on a rate of heating of 100 lb per (sq ft, hr). This rate is high, especially for low-temperature furnaces. As a matter of fact, it is difficult to heat more than 40 lb of steel per (sq ft, hr) to 1200 F. However, a furnace that was sold to be operated at a low temperature is quite often operated at a higher temperature, owing to a change in business conditions. For that reason, the flue sizes figured on the basis of the high rate are none too large for all-round furnaces, and, as before stated, a damper or a tile partly covering the flue will quickly rectify any error due to an excessively large flue.

It must again be emphasized that the values of the table for unit port areas are average figures, for non-regenerative and non-recuperative "in-and-out" furnaces of 40 to 70 sq ft hearth area. In smaller furnaces, the ratio of port area to hearth area must be larger; for large furnaces it can be smaller. The values apply for fuel of high calorific power. For the generation of a given quantity of heat, the products of combustion of producer gas exceed those of rich fuels by 25 per cent. For blast-furnace gas, the excess is 60 per cent. The size of ports and flues must be increased correspondingly. If a fuel that deposits fly-ash is used, the horizontal flues are made even larger, to allow for gradual filling up with ashes, and clean-out doors are provided.

If a large part of the products of combustion escapes through constantly open doors, the flue openings are made smaller than indicated in the tabulation. All the products of combustion from some forge furnaces and bolt-heading furnaces are purposely forced out through the slot in which the heating stock lies.

Rules for general proportions of sizes of vents and flues are not applicable to continuous furnaces. This type of furnace was introduced because it saves labor. So far as fuel economy is concerned, continuous furnaces of all types can be designed to be economical or else to be wasteful. In some continuous furnaces, fuel economy is obtained by heat salvage in recuperators or even in regenerators. Evidently, the multiplicity of design is so great that each type and rate of heating requires a separate calculation. Fuel consumption, rate of flow of products of combustion, and the temperature at which they leave the furnace are determined either by calculation or by comparison with an existing, similar furnace. The calculation is then carried on as given below under "Flues and Stacks."

3. FLUES AND STACKS

The statement was made in the preceding section that gas- or oilfired furnaces need no stacks, but only short flues, as far as their operation is concerned. The products of combustion are discharged from the short flues into the workroom. If their presence is objectionable, a hood, or canopy, with a vent through the roof, is the remedy. Stacks are needed on regenerative furnaces, no matter what the fuel may be (see section 4), and also on many recuperative furnaces.

Coalfired furnaces of any type whatsoever (including furnaces fired with raw producer gas) discharge objectionable, smoky, and sulfurous products of combustion, which are usually delivered through stacks. At the risk of becoming wearisome, the authors wish to state again that there must be a slight pressure in the furnace, and that, in heating-furnace work, the stack has no other function than to carry off the products of combustion. Only a very small fraction of its draft-producing capacity is needed. In consequence, the stack can be of any height, so long as it reaches through the roof and is carried sufficiently high above the surrounding buildings to prevent back drafts or eddies from blowing down into it. The necessity of carrying stacks above the surrounding buildings makes them, with hardly any exceptions, much too high; too much draft is provided, and a damper must be used to counteract the excess draft. On most furnaces, stacks are not only too tall but also too large, for the following reasons: As a protection for the steel shell, a 4½-in. brick lining must be used in stacks serving industrial furnaces. As a rule, bricklayers cannot work in vertical steel shells of less than 36-in. diameter. Occasionally, a 30-in.-diameter stack is found, but it requires a "narrow-gage" bricklayer to line it. In this respect, a stack differs from a horizontal flue. A bricklayer can lie in a flue, and can work himself away from the lining, but the same scheme will not work on a stack. The circumstance that the stack must have a certain minimum height and a certain minimum diameter relieves engineers of the necessity of determining its size, except for large furnaces. For such furnaces, the stack dimensions should be determined by calculation in each individual case. For quick estimates, the following rule may be of value. Make the cross-sectional area of the stack (inside of the lining) equal to approximately 60 per cent of the sum of the exhaust ports or flues, provided that the latter were properly proportioned. This reduction is permissible, first, because the gases cool down on their way, and second, because one large duct of a given cross section offers less frictional resistance than many small ducts of the same cross-sectional area.

In those cases where a stack calculation is required, the method of calculation must necessarily vary with local conditions. The stack may be located at some distance from the furnace; it may serve several furnaces; there may be descending and ascending flues; there may be a waste-heat boiler in the flue; or there may be other variations. It is clearly impossible to consider all these cases in the present volume, but an example will serve to illustrate the general method.

EXAMPLE. It is required to compute the height of the stack that is necessary to produce sufficient draft for the furnace and flue layout shown in Fig. 310, and to check the diameter there shown. The flue lies deep underground, as it has to pass under a foundation. It should be noted that, with this arrangement, a door must be provided at the base of the stack; otherwise the furnace could not be put into normal operation. A fire is lighted at the base of the stack, and draft, which is necessary for starting combustion in the furnace itself, is thus produced.

With regard to draft losses caused by friction, the arrangement may be divided into three parts: (*a*) the two vertical flues 9 by 13½ in. inside, 12 ft long, including the short section at the bottom; (*b*) the 21-by-26 in. (average) flue, 60 ft long, part vertical and part horizontal, and (*c*) the stack itself.

The equations to be used for the draft losses or resistances are those given on pages 369 and 385, with coefficients as in Figs. 300 and 302.

Draft produced by the buoyancy of a vertical column of hot gases, H ft high, at average temperature T F outside air 60 F, barometer 29.92 in., is

$$0.0147H\left[1 - \frac{q_g \times 520}{q_a(T + 460)}\right] \text{ in. of water}$$

q_g/q_a being the ratio of density of the gas to the density of the air. (The coefficient 0.0147 enters the equation on page 367 by substitution of 29.92 in. for barometer and 60 F for temperature of atmosphere.) The probable temperature at various places in the flues must first be determined. This can be done (1) by finding the heat loss through the brick walls of the flue from the chart of Fig. 86, and from the heat-transmitting area of the walls; and (2) by finding the heat lost by the weight of gases that pass through the flue, in dropping from the temperature at the beginning to the assumed (or trial) temperature at the end (or at any other point of the flue). If the two heat quantities are equal, the trial temperature is correct. The temperatures indicated in Fig. 310 have been checked in this way.

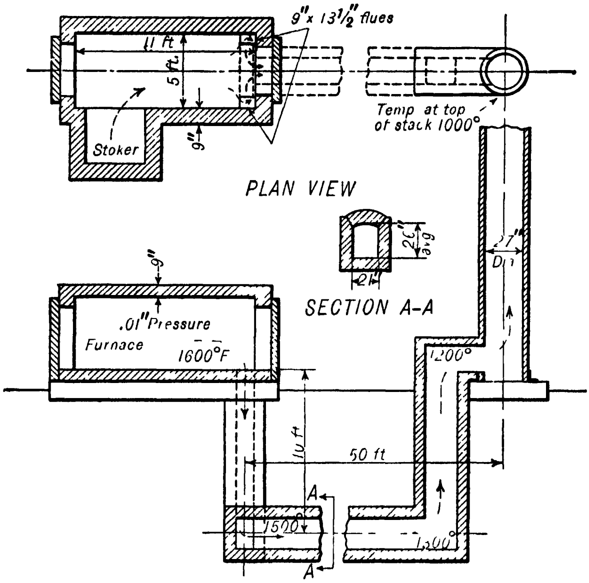

Fig. 310. System of furnace and flues serving as example for stack calculation.

Heat loss from an underground flue, at considerable depth below ground, is not more than one-fourth of that from a flue above ground (unless the ground is wet).

The furnace is coalfired, is used for annealing, and heats 65 lb of steel per (sq ft of hearth area, hr), to 1550 F.

55 sq ft hearth area × 65 lb per (sq ft, hr) = 3575 lb, steel heated per hr

From Fig. 27, the heat required to raise 1 lb of steel from 60 to 1550 F is 253 Btu.

Heat loss from furnace walls, through openings, etc., = 33% of heat in steel. This loss may be determined by the methods of Chapter III.

$$133\% \times 3575 \times 253 = 1{,}203{,}000 \text{ Btu per hr}$$

heat given up by gases in the furnace. With 40 per cent excess air (coal firing), the sensible heat in gases leaving the furnace at 1600 F, from Fig. 110, is

$$4400 + 0.40 \times 3800 = 5920 \text{ Btu per lb of coal}$$

Incomplete combustion loss (1% CO + 1% H_2) = 800 Btu per lb

Lower heating value = 12,128 Btu per lb of coal

$$\text{Fuel consumption} = \frac{1{,}203{,}000}{(12{,}128 - 5920 - 800)} = 222 \text{ lb coal per hr}^*$$

The flue gases consist of 2.61 lb CO_2, 0.45 lb H_2O, 7.34 lb N_2, and 3.81 lb air per lb of coal. At 1600 F, the volume per pound of coal is 717 cu ft, and at 60 F it is 181 cu ft; the density of the flue gas mixture = 0.0199 pcf, or 1.03 times that of air at the same temperature.

(*a*) 9-in. by 13½-in. flues. $L = 12$ ft; B = perimeter = $(9 + 13\tfrac{1}{2}) \times 2 = 45$ in.; $A = 9 \times 13\tfrac{1}{2} = 121\tfrac{1}{2}$ sq in. each

$$\text{Velocity of gases} = \frac{222 \text{ lb coal per hr} \times 717 \text{ cu ft gases per lb coal} \times 144}{60 \times 60 \times 2 \times 121\tfrac{1}{2} \text{ sq in.}}$$
$$= 26.3 \text{ fps at } 1600 \text{ F}$$

Weight of gases per second = 0.880 lb

Average temperature in these flues = 1560 F

$$\text{Average velocity} = 26.3 \times \left(\frac{1560 + 460}{1600 + 460}\right) = 25.8 \text{ fps}$$

$$\text{Friction loss} = 1.03 \times 119.1 \times f \cdot \frac{B}{A} \cdot \frac{Lc^2}{(T + 460)}$$

(The coefficient 119.1 enters the equation on page 369 through the elimination of density q and substitution for diameter in feet of "hydraulic diameter" A/B in inches.)

$$\frac{cD}{u} = \frac{4c \times A}{12u \times B} = \frac{4 \times 25.8 \times 121\tfrac{1}{2} \text{ sq in.}}{12 \times \dfrac{0.00150}{1.03} \times 45 \text{ in.}} = 16{,}000$$

$f = 1.20 \times 10^{-4}$ from Fig. 300, for roughness ratio of 0.016

As the viscosity of 0.00150 applies to air, the density ratio 1.03 is used to correct u for the flue gas mixture.

Pressure drop due to friction

$$= 1.03 \times 119.1 \times 1.20 \times 10^{-4} \times \frac{45}{121\tfrac{1}{2} \text{ sq in.}} \times 12 \text{ ft} \times \frac{(25.8)^2}{2020}$$
$$= 0.0215 \text{ in. water}$$

$$\text{Loss due to velocity head} = 0.184 \times 1.03 \times \frac{(25.8)^2}{2060} = 0.0610 \text{ in. water}$$

Loss caused by two bends

$$= (1.17 + 0.81) \text{ (Fig. 305)} \times 1.65 \text{ (Fig. 303)} \times 0.118 \times 1.03 \times \frac{(25.8)^2}{2060}$$
$$= 0.1284 \text{ in. water}$$

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* On account of increased incomplete combustion loss, the actual fuel consumption is frequently 30 per cent in excess of this value. The flue-gas quantity is then increased by a similar amount.

Draft caused by buoyancy of hot gases in vertical flues

$$= 0.0147 \times 10 \text{ ft} \times \left(1 - 1.03 \times \frac{520}{2020}\right) = 0.1080 \text{ in. water}$$

Pressure in furnace = 0.01 in. water at the hearth

Pressure at bottom of vertical flue

$$= 0.01 - 0.0215 - 0.0610 - 0.1284 - 0.1080$$

$$= -0.3089 \text{ in. water, or actual draft at bottom} = 3/10 \text{ in. water}$$

In this case, the buoyancy of the hot-gas column works against the stack draft, increasing the draft at the bottom of the flue (if measured by a draft gage at that place)

(*b*) 21-in. by 26-in. flue. $L = 50$ ft horizontal + 10 ft vertical = 60 ft; $A = 546$ sq in. $B = 94$ in. Average temperature = 1370 F

Average velocity

$$c = \frac{222 \times 181 \times 144}{60 \times 60 \times 546 \text{ sq in.}} \times \left(\frac{1370 + 460}{60 + 460}\right) = 10.4 \text{ fps}$$

$$\frac{cD}{u} = \frac{1}{3} \times \frac{c}{u} \times \frac{A}{B} = \frac{10.4 \times 546}{3 \times \dfrac{0.00127}{1.03} \times 94} = 16{,}300$$

$$f = 1.20 \times 10^{-4} \text{ from Fig. 300}$$

Friction loss

$$= 1.03 \times 119.1 \times 1.20 \times 10^{-4} \times \frac{94}{546} \times 60 \times \frac{(10.4)^2}{1830} = 0.0090 \text{ in. water}$$

Loss in two bends

$$= 1.03 \times 2 \times 1.08 \text{ (Fig. 305)} \times 1.65 \text{ (Fig. 303)} \times 0.118 \times \frac{(10.4)^2}{1830}$$

$$= 0.0256 \text{ in. water}$$

Buoyancy of vertical column of gas at 1260 F

$$= 0.0147 \times 10\left(1 - 1.03 \times \frac{520}{1720}\right) = 0.1013 \text{ in. water}$$

In this case the buoyancy helps the stack draft, but does not quite counterbalance the hotter column of gases in the 9-in. by 13½-in. flues. Pressure at the base of the stack

$$= -0.3089 - 0.0090 - 0.0256 + 0.1013 = -0.2422 \text{ in. water}$$

or a little less than ¼ in. draft.

(*c*) Stack, 27-in. diameter. $L = H$; $A = 573$ sq in.; $B = 84.7$ in. Average temperature in stack = 1110 F.

Average velocity

$$c = \frac{546 \text{ sq in.}}{573 \text{ sq in.}} \times 10.4 \text{ fps} \times \left(\frac{1110 + 460}{1370 + 460}\right) = 8.51 \text{ fps}$$

$$\frac{cD}{u} = \frac{1}{3} \times \frac{c}{u} \times \frac{A}{B} = \frac{8.51 \times 573}{3 \times \frac{0.0010}{1.03} \times 85} = 19{,}800$$

$$f = 1.24 \times 10^{-4}$$

Velocity at base of stack

$$= \frac{1200 + 460}{1110 + 460} \times 8.51 = 9.0 \text{ fps}$$

Loss in bend at base of stack

$$= 1.5 \text{ (Fig. 302)} \times 0.118 \times \frac{(9.0)^2}{1660} = 0.009$$

$$0.0147\left(1 - 1.03 \times \frac{520}{1570}\right)$$
$$= 0.2422 + 0.009 + 1.03 \times 119.1 \times 1.24 \times 10^{-4} \times \frac{85}{573} \times \mathrm{H} \times \frac{(8.51)^2}{1570}$$

$$0.00969\mathrm{H} = 0.2512 + 0.00010\mathrm{H} \qquad \mathrm{H} = 26.0 \text{ ft}$$

A stack of this height will meet the requirements of the furnace, but a height of at least 40 ft will probably be required to clear surrounding buildings. Consequently, the underground flues could be made smaller, the greater friction loss being compensated by the increased stack draft. This is not advisable, however, because the horizontal flue gradually becomes choked with dust and soot, the free opening becomes smaller, and the friction losses are greatly increased. If the flues are made so small at the beginning that no excess draft is available, then the effect of clogging of the flues cannot be compensated for, and the furnace capacity is much reduced. It will be understood that the inside diameter of the stack cannot be reduced to less than 27 in. for constructional reasons, as explained in the text. A stack damper is needed.

In the above calculation,* account was taken of the difference in density between flue gas and air. The fact that the ratio of densities is so near unity confirms the statement previously made that, as a rule, no account need be taken of the deviation from air density.

4. FLOW OF GASES IN REGENERATIVE FURNACES†

The example of the preceding paragraph prepares, in a measure, for the calculation of flues and stacks of regenerative furnaces, because it considers a flue having ascending and descending branches; but the latter furnaces are more complex because of the variable temperature in the regenerators, their resistance to flow, and the resistance of the reversing valves.

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* For other fuels, the method of calculating diameter and height of stack is the same as that used in the example. It is commendable to provide excess dimensions for the stack and to damper off the excess draft, because the stack does not operate properly when a cold furnace is started.

† In this section, no attention is paid to the influence that the method of arranging the stock, the regularity of reversals, and the method of mixing fuel and air have on flow of gases in the heating chamber and on uniform heating. In practice, these features must, of course, be taken into consideration. Although

At this point, it is expedient to discuss the movement of subdivided streams of a gas, which are either absorbing or giving up heat. The rule is that a gas which is being heated must ascend, whereas a gas which is being cooled must descend, if uniformity of flow and of change of temperature of the different streams is to be attained. For proof of this rule, study Fig. 311.

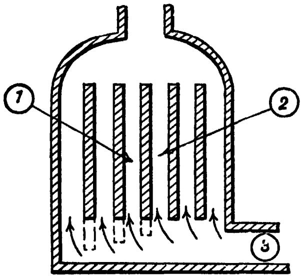

Fig. 311. Flow of a subdivided gas stream.

Let the bricks be hot and let gases that rise in the heater absorb some of the heat. Suppose that one stream, for instance 1, were to flow faster than the rest; then it would absorb more heat and cool the walls of that duct more, and the gas which followed, finding cooler walls, would absorb less heat, thus rectifying matters. If another stream, for instance 2, absorbed heat more slowly than the rest, it would become heavier than the others, less gas would flow through 2, and duct 2 would not be cooled so rapidly as the rest of them, which means that before long its walls would have the same temperature as the others, and the flow would be equalized. If, on the other hand, the gases that absorbed heat were descending, a stream that absorbed too much heat would be hotter than the rest and would flow down very slowly, causing the particular duct in which it flowed to stay hotter than the rest. A stream that did not absorb enough heat, because it passed through a colder duct, would be heavier than the rest, would fall faster, would

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comparatively few industrial furnaces (in the meaning of this book) are of the regenerative type, all steel-melting and glass-melting furnaces are equipped with regenerators. However, regenerator calculations can, with few modifications, be applied to low-lying recuperators. It may again be mentioned that neither regenerators nor recuperators were invented for increasing fuel economy, but for the purpose of obtaining high flame temperature with lean fuels.

cool the duct still more, etc.; in this way the temperature of the various ducts would become very unequal.

It must, however, be understood that upward flow of heat-absorbing gases and downward flow of heat-imparting gases do not necessarily ensure uniform distribution of flow in the different ducts. Inertia of the flowing gas affects the distribution to a considerable extent. If, in Fig. 311, the flue 3 be of small size, the momentum of the incoming gas will carry it toward the extreme left, with the result that the ducts on the left will carry most of the ascending gas. The regenerator will be cold at the left side and hot at the right side. On regenerators that are equipped with peepholes, this uneven temperature distribution can be observed quite plainly. The distribution is made more even by several expedients. The entering flue may be enlarged by so-called "fan tails" (see Fig. 312, which is a plan view). The angle of divergence should not exceed 20 degrees. Baffles may be provided, as shown in dotted lines in Fig. 311. They work well in equalizing the upward flow, but they interfere somewhat with the evenness of the downward flow. A very low entrance velocity, in combination with large chambers above and below the checkerwork, is the best solution.

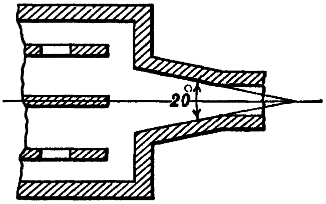

Fig. 312. "Fan-tail" entrance to regenerator.

A thorough investigation of the pressure and velocity distribution in a given regenerator yields very interesting results, but the mathematical difficulties are such as to make it inadvisable to give a detailed account of such an investigation in this volume. One rule, however, may be deduced therefrom, namely, that the flue area, where it joins the regenerator chamber, should not be less than one quarter of the free cross-sectional area of the checkerwork. The authors realize that smaller flues are sometimes used in practice but ascribe that fact to lack of foresight on the part of designers.

Regenerators with horizontal flues are installed every once in a while, but invariably they are, in the end, replaced by checkers with vertical ducts, because there is an ever-present danger that difference in density of hot and cold gases will cause the hot gases to travel along the top and the cold gases to travel along the bottom; heat exchange is, of course, rendered ineffective by such a distribution. The horizontal regenerator should, therefore, not be used for the main checkers. It may be used for auxiliary checkers in horizontal flues which would otherwise have little value as regenerator surface.

Returning to the calculation of flues and ports in regenerative furnaces, we start, just as before, with the determination of the flow of flue gas in unit time. The temperature to which the stock is to be heated is known, and so is the heat per pound of stock at that temperature. A fairly high but still possible rate of heating, in pounds per square foot and hour, is assumed.* The external heat losses of the furnace above the regenerators are computed, and thus the total hourly heat quantity that must be given up by the products of combustion in the furnace is determined. The next step consists either in computing the size of each regenerator from assumed preheating temperature or else in putting under the hearth as large a regenerator as is convenient and computing the temperature of preheat from it. In either case, the curves given in Figs. 154, 155, and 156, Chapter 6, can be used. By reference to the heat-content curves in Figs. 109 to 121, Chapter 5, the hourly flue-gas quantity is readily found for any given fuel.

A diagrammatic view of the passages in a regenerative furnace is given in Fig. 313. On the hearth, the pressure must be atmospheric; hence the stack cannot help to bring air into the furnace (although furnaces are sometimes operated in this way, resulting in excessive air infiltration and scaling of the stock). The calculation of the passages differs, depending on the method of bringing the air into the furnace. If buoyancy of the air in the regenerator and uptake be the sole moving agent, only very feeble pressures are available, and the air-inlet side determines the necessary size of the passages. If a "pusher" fan is available for pumping air into the furnace, fan pressure plus buoyancy, acting on the inlet side, can readily be made to exceed stack draft minus buoyancy, acting on the discharge side, which means that the latter determines the size of the passages.

If the air is moved into the furnace by its own buoyancy only, the regenerators are made as large in cross section as space permits, for the sake of effective heat transmission in them. High velocities and long paths, as obtained in deep single-path checkers or in two-pass checkers, would give better heat transmission per unit of surface, but the draft loss would exceed the draft which is created by buoyancy, which, of course, is impossible. For that reason, the actual air velocity in checkers with buoyancy-actuated flow varies between 1½ and 4 fps, whereas the gas velocity, if producer gas is used, lies between the values of 2 and 3 fps. These velocities are so low that the pressure drop, or draft loss, caused by resistance of the regenerator filling is very small indeed. The pressure drop in solid-chimney checkers can be figured as the sum of the velocity head (page 369) plus friction in rectangular passages (page 371), but, for standard checker setting as in Fig. 173, the drop is not very definitely known. The tests of Kistner* on the experimental regenerator of the German Steelworks Society showed the pressure drop in cold checkerwork to vary exactly as the square of the rate of flow and to be

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* There may be times when uniformity of heating, shape of stock, or kind of material (for instance alloy steel) require a lower rate of heating than the assumed maximum value, but it is advisable to proportion the passages for maximum heating requirements.

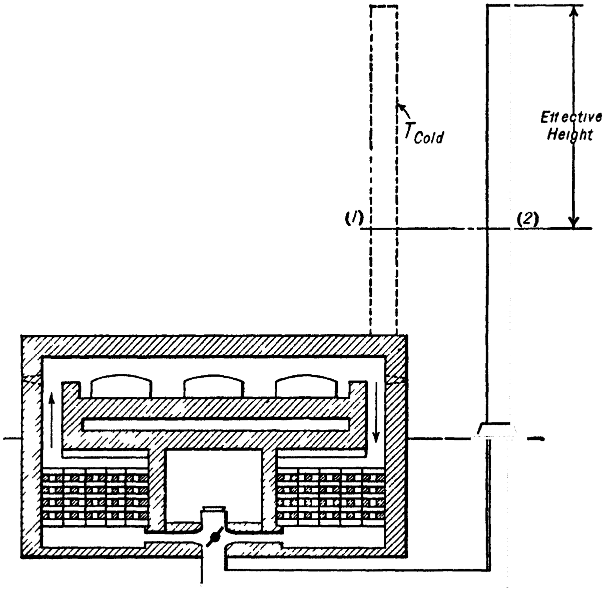

Fig. 313. Diagram illustrating stack effect of different parts of regenerative furnace.

$$p_1 - p_2 = \frac{H(T + 460)c_{60}^2}{2{,}550{,}000\sqrt[4]{D_h}} \text{ in. of water} \qquad (40)$$

in checkerwork H ft high, the average temperature of the air or gases being T F, c_{60} being the velocity in feet per second referred to 60 F, and D_h the hydraulic diameter of the openings in inches (plan view)

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* *Archiv für das Eisenhüttenwesen,* 1930, page 751.

$$= \frac{4A}{B} = \frac{2 \times \text{Width of opening} \times \text{Breadth of opening}}{\text{Width} + \text{Breadth}}$$

This expression applies to checkers with bricks set in line (not staggered). However, since the tests from which this formula was derived were made with cold air and with velocities considerably higher than are usual in checkerwork, and therefore with a much higher Reynolds' number (abscissa of Fig. 300), it is probable that the expression would give results that are too low, when applied to actual checker conditions. Experimental determination of the draft loss in an actually operating regenerator is beset with difficulties, but tests by the senior author on checkers that had been in use for some time indicated that the draft loss may reach values as high as 0.08 in. of water in some cases. (After the checkers have been in use for a time, the openings become partly clogged, and the resistance is much greater than in clean checkers.)

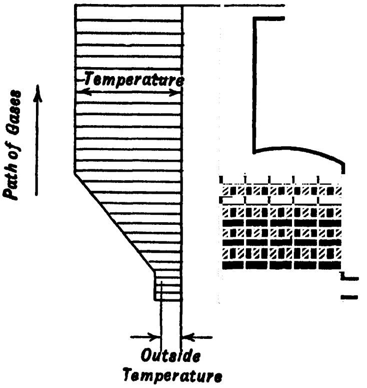

Fig. 314. Temperature distribution in regenerator and uptake.

The total available head caused by the buoyancy can be computed from the height of the ports over the cold-air inlet (see Fig. 314),* proper use being made of the draft diagram (Fig. 295). It ranges from 0.2 to 0.35 in. of water. Allowance should be made for dampering off about one-fifth of this head at the cold-air inlet, when flues and regenerators are clean, for the purpose of maintaining a full flow (by opening the damper) even after flues and regenerators have become partly filled with dust. It is also advisable to use some of the buoyancy head as velocity head for mixing fuel and air at the entrance to the combustion chamber. A velocity of 50 fps is desirable but cannot be had with natural draft, because, at an air temperature of 1500 F, that velocity needs a head of 0.15 in. of water, which, when subtracted from the total available head, does not leave enough for overcoming the friction of flues and valve. The low air velocity interferes with quick combustion, unless burners of proper design are spaced with care, or narrow, alternating ports of air and producer gas are installed (see also Volume II).

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* In the illustration, the temperature is shown to be a linear function of the position of the gas in its travel through the checkerwork. Tests show some variation from this assumption, but it is not great enough to disturb the calculation.

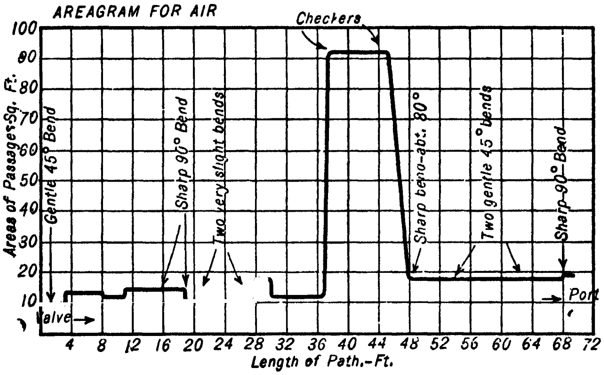


Fig. 315. "Areagram" for air flow in regenerative furnace.

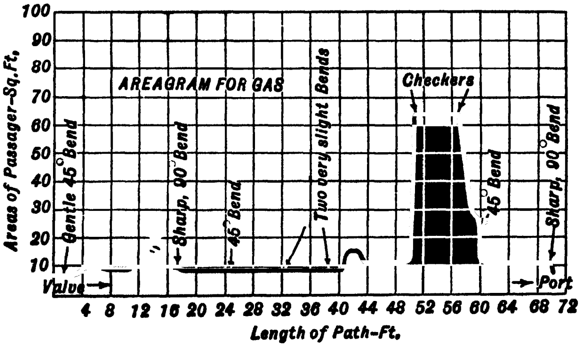


Fig. 316. "Areagram" for gas flow in regenerative furnace.

The remaining head, a scant one-half of the total buoyancy head, is available for overcoming the resistance to flow through the valve, the flue, and the uptake. To guard against unpleasant surprises, some designers plot an "areagram," as shown in Figs. 315 and 316, in which the ordinates are areas, and the abscissae represent length of path of air or gas travel. Bends and their degree of sharpness are noted. From this diagram, the draft loss can be computed. If it exceeds the available head, the passageways must be enlarged, or bends must be eliminated, or both must be done. A glance at Fig. 298 proves that the air velocity in the reversing valve must be kept down to about 15 fps, except for large furnaces with deep checkers.

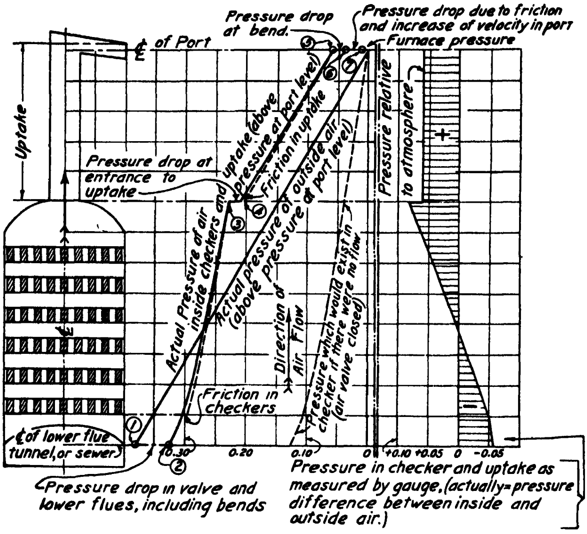

Pressure drop at bend.
Pressure drop due to friction and increase of velocity in port
℄ of Port
Furnace pressure
Uptake
Pressure drop at entrance to uptake
Actual Pressure of air inside checkers and uptake (above pressure at port level)
Friction in uptake
Actual pressure of outside air (above pressure at port level)
Pressure which would exist in checker if there were no flow (air valve closed)
Pressure relative to atmosphere
+
−
Direction of Air Flow
Friction in checkers
℄ of lower flue tunnel, or sewer
①
②
③
④
⑤
⑥
⑦
0.30
0.20
0.10
0
+0.10
+0.05
0
−0.05
Pressure drop in valve and lower flues, including bends
Pressure in checker and uptake as measured by gauge, (actually = pressure difference between inside and outside air.)

Fig. 317. Pressure relations in an air checker.

The authors realize that this method is seldom followed in practice and that most regenerative furnaces are built by comparison with some other furnace which is "working successfully," but they also know that, on furnaces built by this latter method, reversing valves had to be taken out and replaced by larger ones, regenerators had to be changed, new burners had to be installed, and stacks had to be made taller, because they did not produce sufficient draft. In case a regenerative furnace is not working well, a methodical study of the individual draft losses, by means of a differential draft gage, is recommended for locating the places that offer undue resistance.

The question sometimes asked, how it is possible for air to flow from what is apparently a region of low pressure (bottom of checker) to a region of high pressure (uptake and ports), is answered by Fig. 317, which shows that, although these pressures are lower or higher *relative* to the atmospheric pressure at the corresponding levels, the absolute pressure drops steadily in the direction of flow—points 2-3-4-5, etc.

It should be understood that the extreme care here recommended for the proportioning of the passages on the inlet side is not necessary whenever fan draft is available. A pusher fan need produce but little pressure, about 1 or 2 in. of water column, unless the flow of air to the furnace is to be measured by means of an orifice, in which case an additional inch of water pressure is helpful. If the passages are so narrow that more than the above given pressure is needed, then partly preheated air leaks out of the regenerators and makes life miserable for the furnace operators. It is advisable to lock the speed-changing mechanism of the pusher fan; otherwise the heater is everlastingly adjusting the speed of the fan, in hopes of heating the charge at a faster rate.

As regards the outlet side of a regenerative furnace, several matters should be brought to the attention of the inexperienced engineer. First, it must be noted that, although only the before calculated flue-gas quantity passes through the outlet ports of the heating chamber, a much greater quantity goes up the stack. From the outlet port to the stack, the quantity of flue gas grows steadily because of the infiltration of air. The stack produces a partial vacuum which reaches into the regenerator. Not only does air enter through every crack and crevice, but also the bricks themselves are permeable. Reversing valves leak, and stack dampers leak. Excess air amounting to 150 per cent in the stacks of regenerative furnaces is common, and excess air up to 900 per cent is on record. Stacks must, therefore, be proportioned to take care of considerably more than the ideal flue-gas volume. Incidentally, it may be remarked that, on account of the excess air, a low stack temperature is not necessarily an indication of high regenerator efficiency.

The second fact that must be pointed out is the difference of temperature between the gases in the downtake and regenerator on the one hand and in the stack on the other hand. The gases in the downtake and in the regenerator are very light, on account of their high temperature, and, together with the cold air in the imaginary outside branch of the U tube, balance the draft caused by the stack up to the line 1-2 in Fig. 313. Only the stack above that line is effective, and it is at once obvious that regenerative furnaces must, of necessity, be built with tall stacks or with mechanical exhaust means, if they are to operate well. The draft caused by the stack above the line 1-2 must impart velocity to the gases leaving the hearth and entering the ports, and it must overcome the friction of the whole system of passages, bends, regenerator, valve, and stack.

In the design of a regenerative furnace, the problem may be approached in different ways. The height and diameter of the stack may be assumed; the passages, including the valve, may then be calculated so that the assumed stack can draw the gases through them; or else—and this is the better way—the passages and valves may be designed on the basis of conventional velocities, and the stack then calculated to furnish the necessary draft. As a rule, a combination of both methods, including more or less cut-and-try work, is needed. Anyone designing a regenerative furnace for natural draft soon finds out that the designer has but slight leeway and freedom of action. If he makes the passages a trifle too small, the stack becomes extraordinarily high; and, if he makes the stack a little too low, there is not room in the building to accommodate the enormous flues and valves which the small available draft makes necessary. This condition probably accounts for the excessive amount of copying that is done in the "design" of regenerative furnaces. An example will illustrate the methods to be employed in the calculation.

EXAMPLE. A reheating furnace, fired with producer gas, heats 80 lb of steel per (sq ft of hearth area, hr), from 60 to 2200 F. Compute air and gas velocities, pressure drop, and available draft. Are all passages large enough on the basis of 10 per cent excess air? The producer gas comes to the furnace under sufficient pressure to overcome any undue resistance. It is taken for granted that the stack is large enough to provide the required draft to draw the waste gases out of the furnace. The problem in this example is to make sure that the passages are large enough to furnish sufficient air to the furnace. For that reason, only the calculation for the air checker and ducts is given here. In practice, the figures for the producer gas and for the waste gas are calculated by exactly the same methods.

Figure 315 shows an "areagram" of the areas of the air passages, and Fig. 316 shows the corresponding diagram of the gas passages. Hearth area = 570 sq ft.

$$570 \times 80 \text{ lb per sq ft and hr} = 45{,}600 \text{ lb steel heated per hr}$$

From Fig. 27, heat required to raise 1 lb of steel from 60 to 2200 F = 363 Btu

Heat absorbed per hour by steel 363 × 45,600 = 16,553,000 Btu.

Heat losses from the walls, radiation from the openings, etc., as calculated by the methods of Chapter 5 = 4,400,000 Btu per hr.

Heat given up in furnace by products of combustion

$$4{,}400{,}000 + 16{,}553{,}000 = 20{,}953{,}000 \text{ Btu per hr}$$

In this furnace, each air regenerator contains 312 flues, 6 in. by 6 in., each 9 ft 7½ in. long; heating surface = 6010 sq ft. Each gas regenerator has 4160 sq ft of heating surface. Heating surface per 1,000,000 Btu left in furnace (compare Fig. 155).

$$= \frac{6010 \times 1{,}000{,}000}{21{,}153{,}000} = 285 \text{ sq ft}$$

Figure 155 applies directly to recuperators only, but, as explained in Chapter 6, it can be used for regenerators also, by multiplying the surfaces by the factor $\frac{\text{Heat-transmission coefficient in recuperator}}{\text{Heat-transmission coefficient in regenerator}}$, in this case 1.5. Also, the surfaces shown in that figure are minimum surfaces, which are about 80 per cent of actual area required.

$$\frac{285}{1.5} \times 80\% = 153 \text{ sq ft}$$

the equivalent area under the conditions of Fig. 155.

With flue gases entering the air regenerator at 2200 F, and air entering at 60 F, the preheat produced by 153 sq ft of heating surface is shown in Fig. 155 to be 1320 F. The final gas temperature is taken as 1600 F (see discussion in Chapter 6).

From Fig. 116, the heat taken into the furnace by the fuel at 1600 F and the air at 1320 F is

$$31 + 1.10 \times 27 = 60.7 \text{ Btu per cu ft fuel}$$

Heat in products leaving the furnace at 2250 F is

$$101 + 0.10 \times 50 = 106.0 \text{ Btu per cu ft of fuel gas}$$

Lower heating value of producer gas = 137.3 Btu per cu ft

$$137.3 + 60.7 - 106.0 = 92.0 \text{ Btu}$$

heat left in furnaces per cu ft of fuel gas.

Fuel required per hour $= \frac{20{,}953{,}000}{92.0} = 227{,}000$ cu ft (cold measure)

Air required per hour (Fig. 116)

$$= 1.10 \times 1.178 \times 227{,}000$$

$$= 295{,}000 \text{ cu ft (60 F, 30-in. barometer)}$$

Products of combustion per hour:

$$\begin{array}{lll} \text{Air} & 0.10 \times 295{,}000 & \left.\begin{array}{r} 29{,}500 \\ \\ 75{,}000 \\ \\ 340{,}500 \\ \\ 59{,}400 \end{array}\right\} \text{See Fig. 116} \\ CO_2 & \dfrac{\overset{\text{lb } CO_2 \text{ produced per cu ft of fuel}}{0.0382}}{\underset{\text{pcf}CO_2}{0.1156}} \times 227{,}000 \text{ cu ft fuel per hr} & \\ N_2 & \dfrac{0.1110}{0.0737} \times 227{,}000 & \\ H_2O & \dfrac{0.0124}{0.0473} \times 227{,}000 & \\ & & \overline{504{,}400} \text{ referred to 60 F} \\ & & \text{30-in. barometer} \end{array}$$

From the areagram for the gas checkers, the smallest flue area between the valve and the checkers is 7 sq ft, and the temperature is 1100 F.

$$\text{Velocity} = \frac{227{,}000}{7 \times 3600} \times \left(\frac{1100 + 460}{60 + 460}\right) = 27.0 \text{ fps}$$

In the air ducts, the smallest area is 10 sq ft (Fig. 315), and the temperature is 200 F, so that

$$\text{Velocity} = \frac{295{,}000}{10 \times 3600} \times \left(\frac{200 + 460}{60 + 460}\right) = 10.4 \text{ fps}$$

Air checkers. For the purpose of calculating draft required and draft available, the air ducts and flues may be conveniently divided into five sections:

(*a*) Vertical flue, 19 ft long, including valve.
(*b*) Horizontal duct to checker chamber, 11 ft long.
(*c*) Ducts under checkers, 7 ft average length of path of air or gases.
(*d*) Checker flue, 9½ ft long.
(*e*) Vertical flues from checker chamber to furnace ports, 21 ft long.

For each section, calculation must be made for draft required to overcome friction; draft or pressure drop required to produce velocity; positive or negative draft (helping or hindering flow) produced by buoyancy; and loss due to bends.

(*a*) Vertical flue from valve:

$$L = 19 \text{ ft}; \qquad B = \text{perimeter} = 174 \text{ in.}; \qquad T_{\text{avg}} = 140 \text{ F}$$

$$A = \text{average cross-sectional area} = 1580 \text{ sq in.}$$

$$\text{Average velocity of flow} = \frac{295{,}000 \times 144}{1580 \times 3600} \times \left(\frac{140 + 460}{60 + 460}\right) = 8.60 \text{ fps}$$

$$\text{Friction loss} = 119.1 \times f \times \frac{B}{A} \times \frac{Lc^2}{T + 460}$$

(The coefficient 119.1 enters the equation on page 369 through elimination of the density q and substitution for diameter in feet of "hydraulic diameter" A/B in inches.)

$$\frac{cD}{u} \text{ (Fig. 299)} = \frac{4c \times A}{12u \times B} = \frac{4 \times 8.60 \times 1580}{12 \times 0.0002 \times 174} = 130{,}000$$

$$f = 1.1 \times 10^{-4}$$

In Fig. 300, values corresponding to a roughness ratio of about 0.01 will be used. If the brickwork is poorly constructed, with many projecting bricks, or if it is coated with slag waves, the coefficients should be increased accordingly.

$$\text{Friction loss} = 119.1 \times 1.1 \times 10^{-4} \times \frac{174}{1580} \times 19 \times \frac{(8.60)^2}{140 + 460}$$
$$= 0.00337 \text{ in. water}$$

Velocity through valve (maximum)

$$= \frac{295{,}000}{3600 \times 9.62 \text{ sq ft} \times 0.62 \text{ (contraction coefficient)}}$$
$$= 13.7 \text{ fps}$$

Loss due to velocity head $\dfrac{0.118c^2}{T + 460} = 0.0394$ in. water

There is one 45-degree bend and one sharp 90-degree bend. For the 45-degree bend, the loss will be taken as one-half the velocity head; for the 90-degree bend, as equal to 1.65 times the velocity head (Fig. 302).

$$\frac{1}{2} \times \frac{0.118 \times (8.60)^2}{140 + 460} + \frac{0.195 \times (8.60)^2}{140 + 460} = 0.0073 + 0.0240 = 0.0313 \text{ in. water}$$

Negative draft due to buoyancy of hot air in this section

$$= 0.0147 \times 16 \text{ (effective height)} \times \left(1 - \frac{520}{600}\right) = 0.0306 \text{ in. water}$$

(The coefficient 0.0147 enters the equation on page 367 by substitution of 29.92 in. for barometer and 60 F for temperature of atmosphere.)

(*b*) Horizontal duct to checker chamber:

$$L = 11 \text{ ft}; \quad B = 156 \text{ in.}; \quad A = 1440 \text{ sq in.}; \quad T_{\text{avg}} = 170 \text{ F}$$

$$\text{Average velocity of flow} = \frac{295{,}000 \times 144}{1440 \times 3600} \times \left(\frac{170 + 460}{60 + 460}\right) = 9.92 \text{ fps}$$

$$\frac{cD}{u} = \frac{4}{12} \times \frac{9.92 \times 1440}{0.00022 \times 156} = 138{,}000; \quad f = 1.1 \times 10^{-4} \quad \text{(from Fig. 300)}$$

$$\text{Friction loss} = 119.1 \times 1.1 \times 10^{-4} \times \frac{156}{1440} \times 11 \times \frac{(9.92)^2}{630}$$

$$= 0.00244 \text{ in. water}$$

$$\text{Loss in one 90-degree bend} = \frac{0.195 \times (9.92)^2}{630}$$

$$= 0.0303 \text{ in. water}$$

(*c*) Ducts under checkers:

Five ducts, each 15¾ in. by 27 in. by 14 ft. Average length of path of gas travel is 7 ft.

$$L = 7 \text{ ft.} \quad B = 85 \text{ in.} \quad A \text{ (each)} = 425 \text{ sq in.} \quad T_{\text{avg}} = 200 \text{ F.}$$

$$\text{Average velocity} = \frac{1}{2} \times \frac{295{,}000 \times 144}{5 \times 425 \times 3600} \times \left(\frac{200 + 460}{520}\right) = 3.25 \text{ fps}$$

Factor ½ is used because air moves vertically into checkers as well as horizontally, and average value of volume flowing horizontally may be taken as ½ volume at entrance.

$$\frac{cD}{u} = \frac{4}{12} \times \frac{3.52 \times 425}{0.00024 \times 85} = 24{,}400. \quad f = 1.05 \times 10^{-4}$$

$$\text{Friction loss} = 119.1 \times 1.05 \times 10^{-4} \times \frac{85}{425} \times 7 \times \frac{(3.52)^2}{660}$$

$$= 0.000329 \text{ in. water}$$

(*d*) Checker flues:
312 flues, 6 in. by 6 in., 9½ ft long.

$$L = 9\tfrac{1}{2}\text{ ft}; \qquad B = 24\text{ in.}; \qquad A = 36\text{ sq in.}; \qquad T_{\text{avg}} = 800\text{ F}$$

Velocity of flow into checkers = 1.25 fps

$$\text{Velocity head} \qquad = \frac{0.118 \times (1.25)^2}{660} = 0.00028\text{ in. water}$$

For solid-chimney checkers, pressure drop is figured by the aid of Fig. 300.

$$\text{Average velocity in flues} = \frac{295{,}000 \times 144}{(312 \times 36) \times 3600} \times \left(\frac{800 + 460}{520}\right)$$
$$= 2.54\text{ fps}$$

$$\frac{cD}{u} = \frac{4}{12} \times \frac{2.54 \times 36}{0.0007 \times 24} = 1810$$

This value is below the critical value of 3000 (Fig. 300), and the flow is viscous rather than turbulent. $f = 1.02 \times 10^{-4}$.

$$\text{Pressure drop} = 119.1 \times 1.02 \times 10^{-4} \times \frac{24}{36} \times 9\tfrac{1}{2} \times \frac{(2.54)^2}{1260}$$
$$= 0.000393\text{ in. water}$$

Positive draft produced by hot-air column in checkers and in passages at top and bottom. $L = 12$ ft. $T_{\text{avg}} = 800$ F.

$$\text{Draft} = 0.0147 \times 12\left(1 - \frac{520}{800 + 460}\right) = 0.103\text{ in. water}$$

(*e*) Vertical flues from checker chamber to furnace ports:

$$L = 21\text{ ft}; \qquad B = 100\text{ in.}; \qquad A = 627\text{ sq in.}$$

Pressure drop required to produce velocity head at entrance to flues: 4 flues, 27 in. by 23¼ in. Contraction coefficient at entrance = 0.80.

$$\text{Velocity (maximum)} = \frac{295{,}000 \times 144}{4 \times 27 \times 23\tfrac{1}{4} \times 0.80 \times 3600} \times \left(\frac{1320 + 460}{520}\right)$$
$$= 20.1\text{ fps}$$

$$\frac{0.118 \times (20.1)^2}{1860} = 0.0255\text{ in. water}$$

$$\text{Average velocity} \qquad = 0.80 \times 20.1 = 16.08\text{ fps}$$

$$\frac{cD}{u} = \frac{4}{12} \times \frac{16.1 \times 627}{0.0013 \times 100} = 25{,}900. \quad f = 1.05 \times 10^{-5}$$

$$\text{Friction loss} \qquad = 119.1 \times 1.05 \times 10^{-4} \times \frac{100}{627} \times 21 \times \frac{(16.08)^2}{1780}$$
$$= 0.00610\text{ in. water}$$

Sharp 90-degree bend into ports. Velocity = 16.08 fps.

$$\frac{0.195 \times (16.08)^2}{1780} = 0.0283 \text{ in. water}$$

Draft produced by buoyancy of hot air in flue. Effective height = 14 ft.

$$0.0147 \times 14 \times \left(1 - \frac{520}{1860}\right) = 0.148 \text{ in. water}$$

SUMMARY

Section	Pressure Drop or Draft Produced by	Losses, or Negative Draft	Useful Draft
(*a*) Vertical flue from valve to horizontal duct	Friction	0.00337	
	Velocity head	0.03940	
	Bends	0.03130	
	Buoyancy	0.03060	
(*b*) Horizontal duct to checker chamber	Friction	0.00244	
	Bends	0.03030	
(*c*) Ducts under checkers	Friction	0.00033	
(*d*) Checkers	Velocity head	0.00028	
	Friction	0.00039	
	Buoyancy		0.1030
(*e*) Vertical flues from checker chamber to furnace ports	Friction	0.00610	
	Velocity head	0.02550	
	Bends	0.02830	
	Buoyancy		0.1480
(*f*) Furnace pressure		0.0100	
Sum		0.20831	0.2510

The difference, or 0.043 in. of water, is available for overcoming additional resistance caused by choking up of the passages by dust or slag which gradually accumulates in the flues as the operation of the furnace continues, and for velocity head in furnace. When the furnace is first started, the excess draft is taken up by partly closing the air damper; afterward, as the use of this draft becomes necessary for overcoming increased resistances, the damper is opened wider and wider.

The calculation of flues and stacks which serve furnaces with tile recuperators is very similar to that of regenerative furnaces. Such recuperative furnaces are illustrated in Figs. 318 and 69. The horizontal zigzag flues through which the products of combustion flow in the recuperator offer resistance to flow, and so do the underground flue and the stack. The buoyancy of the hot flue gases in the recuperator counterbalances the smaller buoyancy of the colder gases in a longer (vertical) section of the stack. A study of the previously given example will facilitate the calculation of passages, flues, and stack for a furnace equipped with a tile recuperator. If such a furnace is of the

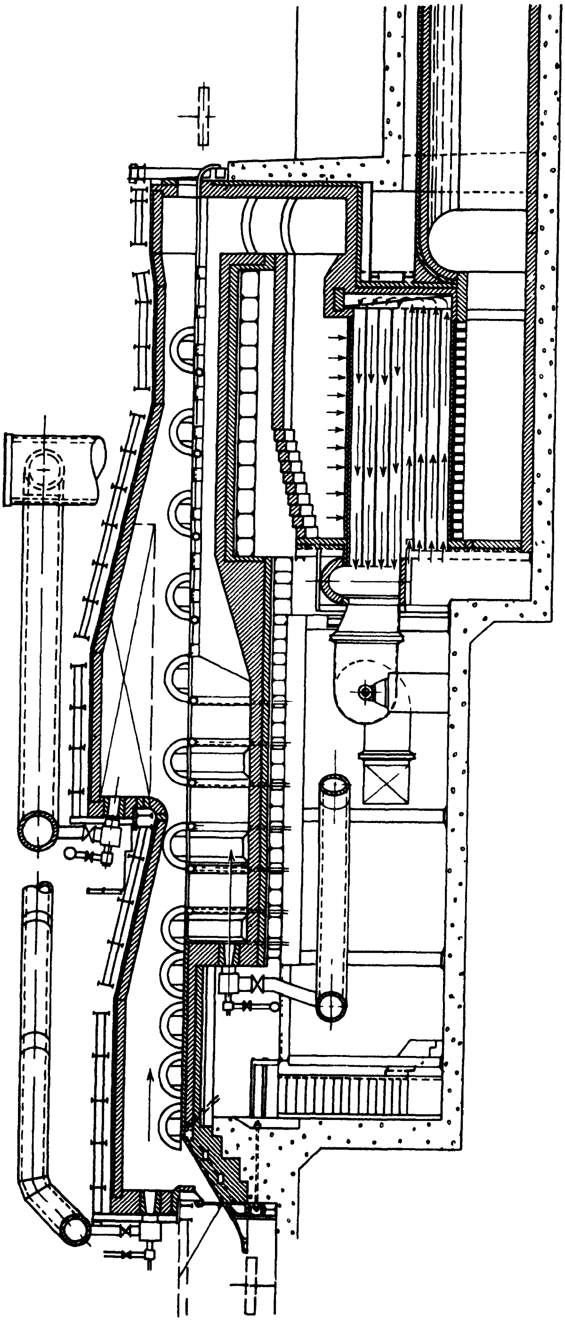


Fig. 318. Furnace with low-lying recuperator.

continuous type, an uncertainty enters, which is caused by inward or outward leakage at the charging end. At that end, some clearance must exist between bottom of door and top of charge. And furthermore, the stock is never so wide as the door opening. As previously explained, the slightest pressure at the discharge end (referring to gases) of a furnace causes large volumes of flame and smoke to billow out from under the constantly open door; and the slightest vacuum in the furnace causes cold air to enter the flue-gas ducts of the recuperator. The latter happening is the more harmful of the two. Not only is the temperature of the products of combustion reduced by being diluted with cold air, but their volume is increased, whereby, in turn, frictional resistance is increased. For that reason, a very slight pressure is maintained at the charging end of a furnace. If the furnace is operated in that manner, passages, flues, and stack designed to take care of the theoretical volume of flue gases are adequate, except in the not infrequent event that the owner decides later on to heat a much greater hourly weight in the same furnace. In that case, a new and larger stack is needed, or else a mechanical device must be installed for exhausting the flue gases.

Ceramic (tile) recuperators invariably become leaky under the effect of contraction and expansion of week-end shutdowns. For that reason an effort is made so to operate the recuperator that the pressures on the flue-gas side and on the air side are equal. And, since the stack causes a slight vacuum, the air should arrive with atmospheric pressure at the top of the recuperator. The air pressure that is needed at the burners is generated by a hot fan between recuperator and burners. The fan is preferably equipped with a butterfly damper on the pressure side.

Mechanical devices for moving gases are needed less at the outgoing end of a regenerative furnace than at the ingoing end, because the stack can always be made large and tall enough to pull the flue gases through the valves, the checkers, and the passages. Nevertheless, mechanical devices are in use on some industrial furnaces, for instance on pit furnaces (soaking pits). Such devices are either direct-acting suction fans which generate a pressure difference of a few inches of water column or else high-pressure fans which aspirate the spent gases by means of an air jet. The fan air passes through a nozzle which is set ahead of the throat of a Venturi tube, as shown in Fig. 319. Fans through which the products of combustion flow after they have passed through regenerators need not be made of heat-resisting metal, unless the regenerators are too small. This statement applies to industrial furnaces in the sense of the present book, but not to steel-melting furnaces, from which the gases are discharged at a much higher temperature. However, the bearings of discharge fans must be water-cooled in any event.

In Fig. 319, only the checker and short stack on one side of the furnace are shown. If a damper in the stack be closed, the air issuing from the jet must reverse its direction and flow through the regenerator. If the damper be open, the jet air aspirates the products of combustion and ejects them through the Venturi stack. The dampers are the reversing valves.

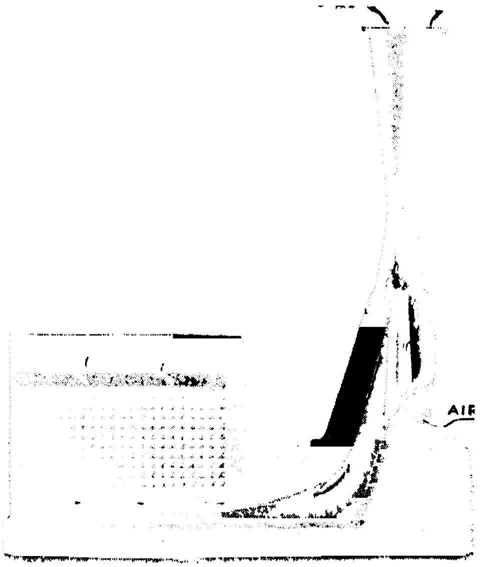


Fig. 319. Isley system for control of reversing furnaces.

Fan-air operated jets and even steam jets have been put into stacks as an afterthought in order to increase the draft. A steam jet pointing upward at the bottom of an unrestricted stack indeed increases the draft. However, the small increase in draft is bought at the expense of an everlasting, very high steam consumption. The principle is fundamentally wrong.

If a fan-actuated air jet is placed in an existing stack, its proper place is near the top of the stack, as diagrammatically indicated in Fig. 320. Such jets have been placed near the bottom of the stack. However, that location is wrong, for two reasons: The hourly weight of the ejection air ranges between 60 and 100 per cent of the hourly weight of the furnace gases. If that quantity of cold air enters at the foot of the stack, the stack gases are chilled and the natural draft is reduced (see Fig. 295). The ejector air also greatly increases the volume flowing through the stack in unit time, whereby the frictional resistance to flow is greatly increased. Cases are on record in which the "draft-increasing device" actually reduced the available draft. The theory of aspiration by jets is not yet fully developed, in spite of the expenditure of much thought and effort. It is advisable to procure jet-aspiration equipment from those who have made exhaustive (in more senses than one) tests with their apparatus. The senior author went deeply into the theory of jet induction (*Heat Treating and Forging,* January to March 1932) and hoped to substantiate his findings by a series of carefully planned tests with jets which discharged into active stacks. However, the tests showed that seemingly insignificant changes in the proportions produced great differences in the results. For that reason, results of the tests are not given here.

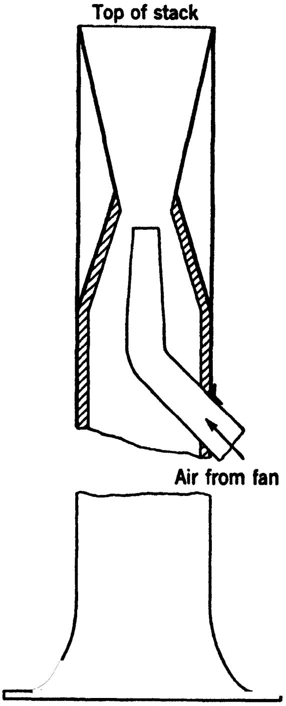


Fig. 320. Increase of draft by air jet.

Flow of Gases through Tube Banks. Recuperators of several types consist of tube banks across which either air or products of combustion flow. The resistance to flow through tube banks depends on a large number of variables. Equations for that resistance have been derived on the basis of the test data. These equations are necessarily complicated. For that reason, it is preferred to refer to the original publications. See Chilton and Genereaux, *Transactions American Institute of Chemical Engineers,* 1933, pages 161 to 173; and three papers in the *Transactions American Society of Mechanical Engineers,* 1937: E. C. Huge, pages 57 to 81; O. L. Pierson, pages 563 to 572; and E. D. Grimison, pages 583 to 594. "Heat Transmission" by McAdams and "Der Industrielle Wärmeübergang" by Schack may also be consulted.

5. CIRCULATION OF FURNACE GASES AND MEANS FOR PRODUCING IT

A distinction exists between flow of gases and circulation of gases. Products of combustion may flow directly from the burner (or inlet port) to the vent, or they may be compelled to pass the same spot twice or more times in succession, in which latter case they truly circulate. In this section the term circulation will be used rather loosely and will be applied both to single flow and to truly circulatory flow.

In the section on heat transfer, the variation of the ratio (heat transfer by radiation to heat transfer by convection) with temperature was discussed. From that ratio it follows that rapid and uniform heating is obtained at high temperatures (1800 to 2300 F), if every square inch of the surface of the charge can see (figuratively speaking) thick, glowing gas streams and the hot walls, without necessarily coming in contact with the hot gases. For reasons that are explained in Volume II, oxidation of the charge is smallest if there is no actual contact. Most shapes of material that is to be heated to a high temperature lend themselves to being heated principally by radiation.

At low temperatures (1000 to 1300 F), radiation is so weak that convection becomes an important factor in heat transfer. In heat transfer by convection, the hot gases must actually contact the charge and pass over or through it, preferably several times in succession. The conclusion to be drawn from this reasoning is not that circulation is undesirable at high furnace temperatures, but that it is positively necessary at low temperatures if quick and uniform heating is to be obtained.

From Chapter 1 it follows that there is an endless variety in the shapes of furnaces, in arrangement of the stock within them, and of the heat-liberating devices. For that reason, it is impossible in the present volume to trace the circulation of gases in every type of heating furnace and annealing furnace and for every conceivable arrangement of the charge. Only general principles can be studied here; their applications to individual furnaces are dealt with in Volume II.

There are five operating causes for circulation of gases in furnaces:

1. Change of volume during a change of temperature.
2. Difference in density between hot and cold gases.
3. Kinetic energy of flowing gases due to pressure of incoming fuel and air.
4. Furnace pressure in conjunction with draft in flues.
5. Effect of wind acting on outside of furnace.

Means for giving direction to the circulation are as follows:

6. Location of heating and combustion devices.
7. Direction of jets of hot gas or flame.
8. Location of outlet ports.
9. Location and slope of walls, roofs, and baffles.
10. Arrangement of heating stock in furnace.
11. Fans.

A discussion of each of these propositions, together with a few illustrative examples, will be of assistance.

(a) Change of Volume Due to Change of Temperature. The volume of a given weight of a gas is proportional to the absolute temperature, if the pressure remains constant; and it remains practically so in furnace work. This agency, taken by itself, causes but little of the gas circulation in furnaces, but it certainly increases and intensifies the action of cause 3. The outflow of hot gases that occurs when an electric furnace is being heated up is almost entirely due to increase in volume with increase in temperature. The increase in volume during combustion is very helpful in filling combustion spaces and ducts with heat and flame and in causing an even flow of gases through the heating chamber, although the fuel and air mixture may come into the furnace in separate jets.

(b) Difference in Density between Hot and Cold Gases. This difference produces the well-known stack effect. It is very beneficial if properly used, but is also harmful if allowed to act in certain ways. It is beneficial because it brings the air into the furnace through low-lying regenerators or recuperators. It brings products of combustion into the heating chamber from low-lying coal fires. It is practically the only agency that produces circulation inside of muffles and electrically heated furnaces. It causes heat and flame to seek the roof of the furnace and to pass above the heating stock, which lies in a colder atmosphere. If quick and uniform heating, at temperatures of 1000 to 1400 F, is to be attained, this tendency must be counteracted by other means, which are described below. The difference in density between hot and cold gases tends to produce an undesirable circulation (as shown in Fig. 321), allowing cold air to enter at the bottom of the door, or through cracks in the brickwork. It is the desire to avoid this condition that has led to the adoption of the practice of carrying just a little more than atmospheric pressure in the furnace at hearth level. Of course, this expedient does not overcome the circulation tendency due to difference in density. It does stop the entrance of cold air, but it causes more heat to be lost through openings around the top of the door.

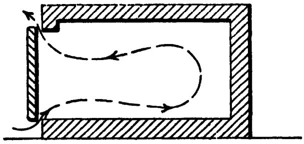


Fig. 321. Undesirable circulation caused by stack effect of hot furnace.

Where it is especially important to stop the entrance of atmospheric air, nozzles may be located just below the bottom of the door, directing high-velocity jets of air away from the furnace. The circulation shown in Fig. 321 can thereby be stopped or even reversed; the latter condition, however, can be avoided by adjusting the air pressure at the nozzles.

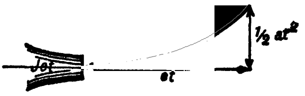
In an end-discharge continuous furnace (such as shown in Fig. 318), on account of the stack effect due to the slope of the hearth and poor fit of the discharge door sections, the leaking in of cold air is especially serious. It not only reduces the economy but also causes increased temperature differences in the billets, reduces the heating rate, and results in excessive scaling. Well-fitted mechanically operated doors (shown in Fig. 263) eliminate air leakage to a great extent. "Conditioning burners" have been used to provide an excess of gas near the door and over the billets.

If a hot gas stream of density q_1 is entering horizontally with velocity c into a cooler medium of density q, its buoyancy makes it rise. The upward force acting on a volume V equals $V(q - q_1)$; the mass of that volume is Vq_1/g, and the upward acceleration

$$\left(\text{as expressed by } \frac{\text{Force}}{\text{Mass}}\right) \text{ equals } \frac{(q - q_1)g}{q_1} = g\left(\frac{q}{q_1} - 1\right) = a$$

The shape of the gas stream is an upward-bent parabola, points of which are found by this simple reasoning. After time t sec, the horizontal component of the path is ct, while the vertical component is $\frac{1}{2}at^2$ (see Fig. 322). However, calculations of this sort must be used with great care. In the first place, the difference of density between the entering stream and the surrounding medium is seldom accurately known; and, in the second place, viscous drag causes entrainment and diffusion to such an extent that all differences of density are obliterated after some distance of travel.

Fig. 322. Rising stream of hot gases.



(c) Kinetic Energy of Flowing Gases Due to Pressure of Incoming Fuel and Air. The fact that air and gas, or oil, or powdered coal are brought into the furnace with considerable velocity (which is necessary to produce both mixing and pressure at the level of the hearth) furnishes one of the most valuable means of effecting a lively circulation in the furnace, provided that the latter be so shaped as properly to direct the gases and to maintain the circulation (see below, under **i**). A very small stream of gas can maintain a perpetual circulation in a circular chamber with a central outlet (as shown in Fig. 323). A simple case like this one is amenable to mathematical treatment, but very little is gained thereby, because the "Rotaflame" furnace is only one special case out of a great many, and because, in all cases of low-temperature heating, circulation through and between pieces of the charge is the great desideratum.

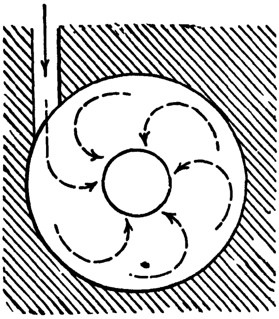


Fig. 323. Circulation maintained by a small jet.

In those furnaces in which circulation is most necessary, that is to say, in furnaces carrying temperatures between 1000 and 1400 F, it is out of the question to let the flame or the extremely hot products of combustion touch the stock; they must be diluted and mixed with products of combustion which have been in the furnace for some time. This is often done by induction or entraining. Many attempts to secure furnace circulation by this method have been failures, because the designers did not realize that length is needed for induction. In section 1 of this chapter, the laws of induction were given; in practice, they are complicated by the fact that induction and combustion go on simultaneously and that the density of the jet differs from the density of the surrounding medium. For these reasons, the following example does not give quite the true answer, but it is exact enough to impress on furnace engineers the relation that must exist for proper induction.

Example. Products of combustion with a temperature of 2400 F flow in a jet of 6-in. diameter with a velocity of 150 fps. They induce other products of combustion, having a temperature of 1200 F. How far from the mouth of the jet will the temperature of the jet have dropped to 1400 F?

For the sake of simplicity, take the specific heat to be constant. Then calling W_1 the weight flowing from nozzle per second and W_2 the weight entrained, we have

$$2400W_1 + 1200W_2 = 1400(W_1 + W_2)$$

or

$$\frac{W_2}{W_1} = \frac{1000}{200} = 5$$

From the momentum formula, velocity at section of mixing,

$$c = \frac{150}{\left(1 + \frac{W_2}{W_1}\right)} = \frac{150}{6} = 25 \text{ fps}$$

From the fact that both for the mouth and for the mixing section, weight in unit time equals area times density times velocity, follows

$$A_2 = A_1 \cdot \left(\frac{c_1}{c_2}\right)^2 \cdot \frac{v_2}{v_1}$$

A denoting area, v specific volume, and c velocity.

By substitution,

$$A_2 = A_1 \times 6 \times 6 \times 0.58 = 21A_1$$

But

$$A_2 = \pi(3 + L \tan j)^2 \quad \text{and} \quad A_1 = \pi(3)^2 \qquad \text{(see Fig. 324)}$$

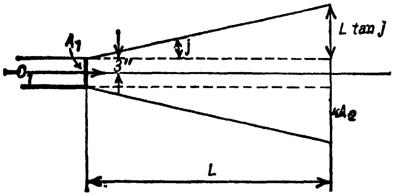


Fig. 324. Diagram illustrating entraining action of a jet.

Hence,

$$21 = \left(1 + \frac{L}{3}\tan j\right)^2, \quad \frac{L}{3}\tan j = \sqrt{21} - 1, \quad L = \frac{3}{\tan j}(\sqrt{21} - 1),$$

$$j = \tfrac{1}{2} \times 25^\circ \quad \text{(see page 379)}$$

$$L = \frac{3}{0.22}(4.6 - 1) = 49 \text{ in.}$$

Evidently, the length that is needed for induction is proportional to the diameter of the jet. If, in a given design, the necessary length is not available, change the design, or use several jets of smaller diameter.

It must again be emphasized that this calculation is only an approximation. The velocity is not uniform in the expanded jet, being high in the center and practically zero at the edge. In accordance with this distribution, the temperature is low at the edge and high in the core.

Radiation of heat from the products of combustion tends to reduce the temperature in a shorter length, while after-burning retards the drop of temperature.

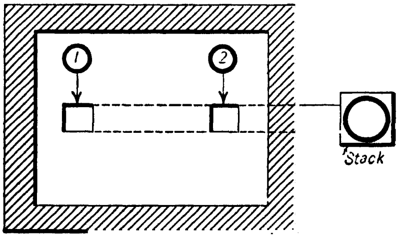


Fig. 325. Wrong flue proportions.

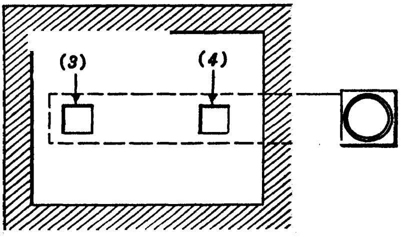


Fig. 326. Right flue proportions.

(d) Furnace Pressure in Conjunction with Draft in Flues. The action of pressure difference is so self-evident as to call for little discussion. Fluids always flow from regions of high pressure to regions of low pressure (unless the pressure difference is caused by the weight of an overlying column of fluid). Greater pressure differences produce faster flow (for a given density of fluid). This latter proposition is so simple that every furnace engineer might be expected to heed it, and yet mistakes are made by neglecting it.

Figures 325 (showing the wrong way) and 326 (showing the right way) illustrate the point in question. In Fig. 325, more flue gas will be drawn through port 2 than through port 1, because the pressure drop through port 1 plus flue to a point under port 2, must be the same as the pressure drop through port 2 alone, and that equality of pressure drop can be had only by passing more flue gas through port 2 than flows through port 1. If, on the other hand, the flue is made quite large (Fig. 326) compared to the ports 3 and 4, the pressure drop and the flow through both will be practically equal.

(e) Effect of Wind. The majority of industrial furnaces are not only located indoors, but are also clad in sheet steel. For both reasons, wind has little effect on circulation and pressure in such furnaces. Occasionally, with a combination of strong wind and windows, ventilators and doors wide open, an effect is noticed in the action of inspirating burners. Matters are different with furnaces that are located in the open, for instance, by-product coke ovens.

(f) Location of Heating and Combustion Devices. With coal on the grate, the choice of location is somewhat limited, and baffles, etc., as are described under method **i** are necessary to direct heat and circulation in the desired channels. With powdered coal, oil, or gas, greater freedom exists.

Furnaces may be endfired; they may be sidefired. Burners may be located above the charge or under the hearth. The number of burners may be varied. Resistors and radiant tubes may be located above or below the charge. The multiplicity of arrangement is too great to permit a detailed discussion.

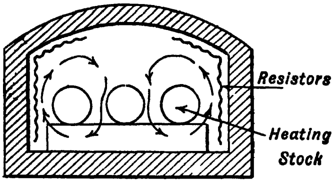


Fig. 327. Convection currents induced by hot resistors.

In some furnaces that are heated by resistors or by radiant tubes, circulation is obtained by mechanical means. Without such means, a sluggish circulation exists in such furnaces as indicated by the arrows in Fig. 327 which diagrammatically represents an electrically heated furnace.

(g) Direction of Jets of Hot Gases or Flame. In comparatively few cases will satisfactory operation result from directing flames into and between the pieces that are being heated. There is danger that the tongues of flame will overheat some of the pieces and that excessive scaling will take place, because a flame is a sign of incomplete combustion. As a rule, it is better to direct the hot gases along a brick wall and to deflect them so that they are bent back into the pile of stock (or around it) (as indicated in Fig. 307), or else to induce a current by viscous drag, which method was indicated under **c**. What occurs when the flames are sent directly into the charge is illustrated by Fig. 328, which shows a perspective view of the hearth and ports of a regenerative annealing furnace, with two trays of castings on the hearth. The hearth is shown as it would appear with the front wall removed. When the furnace is in use, the whole hearth is covered with trays. Air and gas pass out of the ports and are directed straight into the pile of castings. If a tray of cold castings is set close to the ports, flame can be seen passing between the castings and even beyond the tray. As long as the castings remain fairly cold, no harm can result, either from local overheating or from scaling. When the castings become red hot, scaling and non-uniform heating are unavoidable, unless the trays are moved to some distance from the ports; then every stream of gas and air induces enough of the furnace atmosphere to bring the so-called flame temperature down; besides, combustion is so nearly completed that scaling is low, and, finally, the burning gases have given up a considerable amount of heat by radiating to the walls and the roof, before they contact the charge. In some works, the castings are left close to the ports until taken out, and the scaling is accepted as a "necessary evil." *

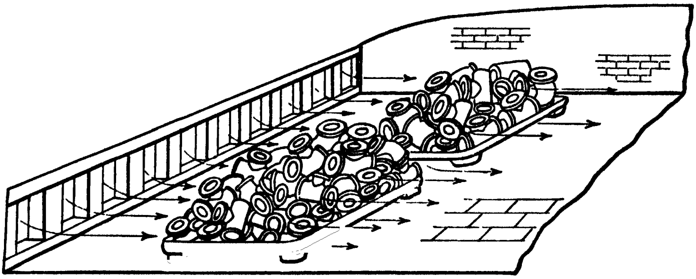
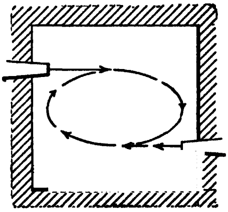


Fig. 328. Wrong location of charge in an annealing furnace.

Fig. 329. Circulation produced by offset burners.



Offset jets, firing in opposite directions, as indicated in Fig. 329

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* It is needless to say that the practice described in this paragraph is very poor. The saving effected by the regenerative furnace is wholly imaginary in this case, because some of the charge must be heated to a temperature that is much higher than is required for annealing, in order to bring the cooler part of the charge up to annealing temperature.

produce a lively circulation. This illustration may be taken to be a plan view or an elevation. In the latter case, the heating stock must be properly blocked up so as not to interfere with the circulation.

Circulation of this type is very effective in low-temperature furnaces, in which little danger of scaling exists. The flame is tempered by excess air that is injected at high velocity.

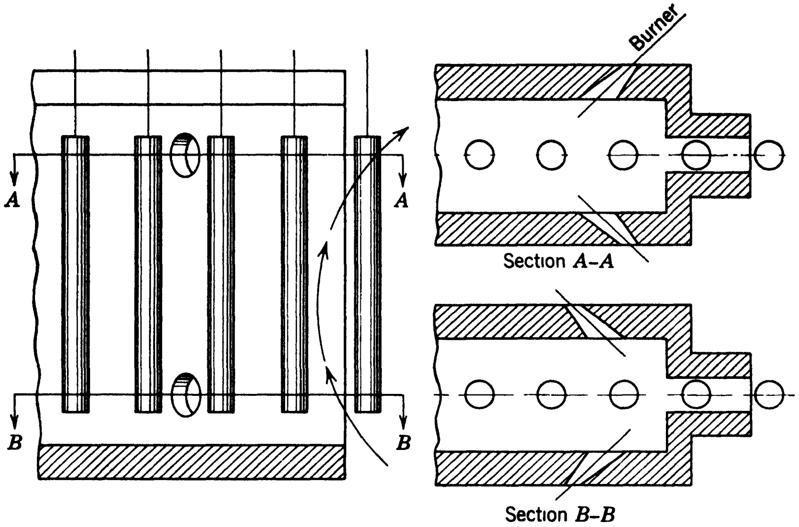

Fig. 330. Device for reducing air infiltration in tall furnaces.

Jets may also be used to prevent circulation. If long, hanging objects, such as pipe, are carried through a continuous furnace, as shown in Fig. 330, buoyancy causes a circulation which is indicated by arrows. The top is overheated, and the bottom is too cold. If the burners are arranged as shown in the plan view, their jet actions counteract the undesirable circulation that is produced by buoyancy. When the furnace is cold, the bottom burners throw flame out of the furnace into the workroom.

The action of jets in underfired furnaces was explained on page 382.

(h) Location of Outlet Ports. Flue gases must go to the outlet ports; if these ports are correctly located and have the proper relative sizes, the gases can be drawn to any point in the roof, sidewalls, or hearth. In the location of ports, judgment must be exercised to avoid short-circuiting of the gases from the burners or inlet ports to the outlet ports. In certain types of annealing furnaces, it has been found very advantageous to have many small ports uniformly distributed in the hearth, because that arrangement carries the hot products of combustion to every part of the furnace. Examples will be given later

(i) Location and Slope of Walls, Roofs, and Baffles. Circulation is helped or hindered by proper or improper location of walls. Furnace gases have mass, inertia, and friction. In many respects, their flow is similar to that of water and is affected by walls in a corresponding manner. A few examples will illustrate their effect.

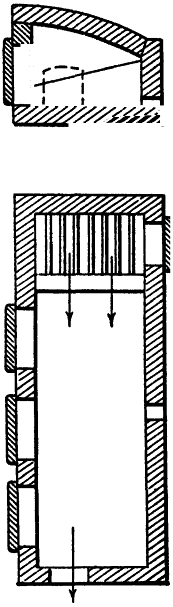

Fig. 331. Arrangement of bridge wall, roof, and port for uniform heating.

Although present economic conditions in the fuel industry of the United States have caused the firing of industrial furnaces by coal on the grate to become rather infrequent, coalfired industrial furnaces are common in some parts of Canada and in many European countries. For that reason, the remarks in the following paragraph are appropriate.

In endfired furnaces, it is difficult to heat the stock lying close to the doors, unless so much pressure is carried in the furnace that hot gases are forced out around the doors, or else the flame is thrown toward the doors, or else a greater quantity of gas is made to pass along the doors. This last solution is by far the best, but is, unfortunately, not found in practice so often as it should be. Three of the methods by which greater gas flow along the door side is obtained are shown in Fig. 331, which represents a coalfired furnace. First, the roof is drawn down deep on the rear end, much lower than on the door side; second, the bridge wall is lower on the door side than on the rear end; and third, the outlet flue is offset so as to be closer to the door side. This last feature is an application of the principle expressed under **h.**

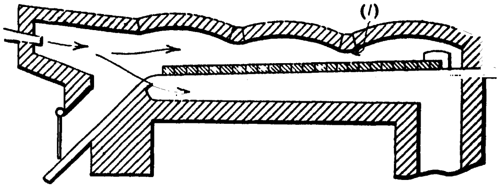

Fig. 332. Circulation in continuous furnace as affected by shape and location of roof.

Another example of the effects of walls and baffles on circulation is given by Fig. 332, which represents a continuous billet-heating furnace with split flame. The wavy roof * mixes the gases and deflects hot gases down upon the billets, and the restriction caused by the dip at point 1 compels part of the heating gases to travel under the steel.

It may be well to mention here the Yesmann "inverted weir" theory of flow of gases in furnaces.† Starting from the hydraulic analogy of the experiments with kerosene and water made by Groume-Grjimailo to simulate the flow of hot and cold gases, Yesmann concluded that flow under a "knuckle" in a furnace roof or under a curtain wall, and with certain modifications the flow under any furnace roof, is equivalent to flow over an inverted weir, and that the thickness of the moving gas stream would be

$$H = k\sqrt[3]{\frac{Q^2}{b^2(T + 460)}}\ \text{ft}$$

where the volume of gases flowing at T F is Q cfs, the width of passage is b ft, and factor k ranges from 2.4 to 2.9, varying with b and H. If the height of passage under the contraction exceeded the height H as figured from the above expression, he assumed that the gas stream would not fill the passage and that the gases would "channel" under the roof, leaving a cold stagnant layer on the hearth. Although chemically dissimilar fluids, such as light oil and water, of different densities and having relative motion to each other, exhibit a well-marked boundary between the lighter top fluid and the heavier bottom fluid, it is known from modern developments of the theory of flow of fluids that no such thing as a lower free boundary can exist in the case of identical fluids such as hot and cold gases; on the contrary, the viscous drag of the gases under the roof will either carry along the gases near the hearth or will at least cause recirculation of these gases. Although the Yesmann theory can be applied with some degree of justification to tall furnace chambers, its field of use is limited to very slow rates of flow, and the attempt to extend it to all cases of flow of gases in furnaces is wrong and misleading.

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* In spite of the advantages in circulation obtained by the multi-arched, wide roof, that type of roof is being replaced by a suspended flat roof. The arch at the hot end of Fig. 332 burns thin, collapses and causes collapse of the whole roof.

† Groume-Grjimailo-Williams, "Flow of Gases in Furnaces," 1923, pages 40 and following.

The following experiment made by the senior author is of interest in this connection. Hot smoky gases were admitted at a very low rate of flow under the roof of a full-sized furnace (as shown in Fig. 333*a*).

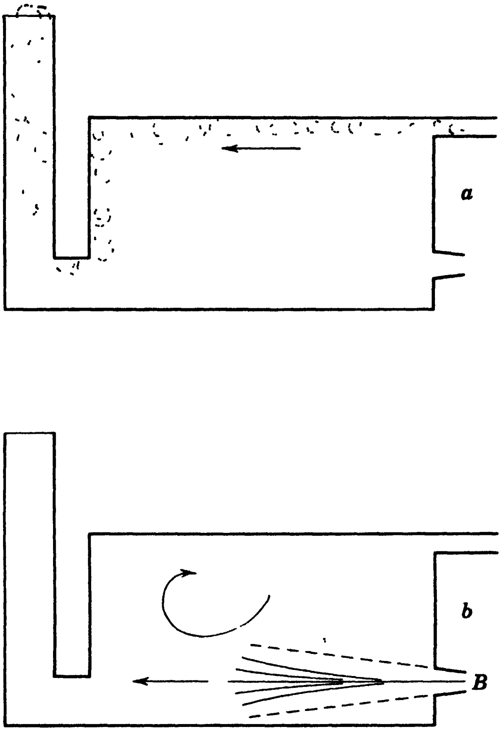

Fig. 333. Effect of jets on circulation.

They rolled along under the roof and passed out at the vent as shown in the *a* part of the illustration. But, when the oil burner *B* was started, the hot smoky gases were sucked into the path of the oil flame (as indicated in Fig. 333*b*) and no longer followed the roof. The same effect was produced when only the atomizing steam, but no oil, was admitted.

(j) Arrangement of Heating Stock in Furnace. Whereas principles **a** to **i** concern the furnace builder, this one must be observed by the user. Violation of the rule that the stock must be placed to allow circulation is very common. In furnaces that are built with circular cross section for the express purpose of securing the best possible circulation, ingots are often placed on the floor (as shown in Fig. 334, left)* whereas they should be placed on bricks or blocks at each end, as shown at the right of the same illustration. In other furnaces, castings are piled promiscuously and obstruct outlet ports in the hearth or sidewalls. Bars or billets are placed so close together that gases cannot pass between them, and many other errors are made. By observing a few ordinary precautions in the placing of the stock, the heater can improve the uniformity of heating, reduce the time that is required, and save fuel.

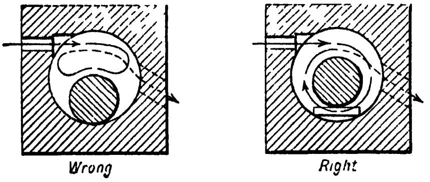

Fig. 334. Methods of placing stock in furnace.

A furnace in which the effects of all the factors discussed in the preceding paragraph are clearly evident is the one-way-fired pit furnace illustrated by Fig. 335. The heater places the ingots far enough apart to permit circulation of hot, radiant gases between the ingots. At high fire, the gases follow the path 1–2–4. If the burner is correctly designed, combustion is almost completed when the burning gases have traveled the length of the pit. Reversal of direction of flow at the wall causes mixing and completes the combustion. Ingot No. 5 is well heated. Ingot No. 6 is relatively cold, especially at the bottom. That ingot is brought up to temperature, if the burner is turned down, because in that case, the gases take the path 1–3–4. Uniformity of temperature throughout the pit is obtained either by alternating high fire and low fire or by high fire during the heating period and by the low fire during the soaking period. In a third method of circulation, the pit is fired full blast until the hottest ingot is slightly too hot. The fuel is then turned off, while the flow of air is continued. With rich fuels such as oil or natural gas, this method of circulating produces good uniformity of temperature. It should be mentioned that rapid heating damages ingots of certain alloy steels, unless the ingots had been slowly preheated in a separate heating chamber.

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* Although the circular furnace has been selected to illustrate excellent circulation, it must be realized that it is not necessarily an example of uniform heating. As a rule, the heat inlet is placed near the door. Instead of having a cold spot near the door, this furnace has a hot zone there. If uniform heating is desired, an auxiliary burner about halfway between the main inlet and the outlet port should be used.

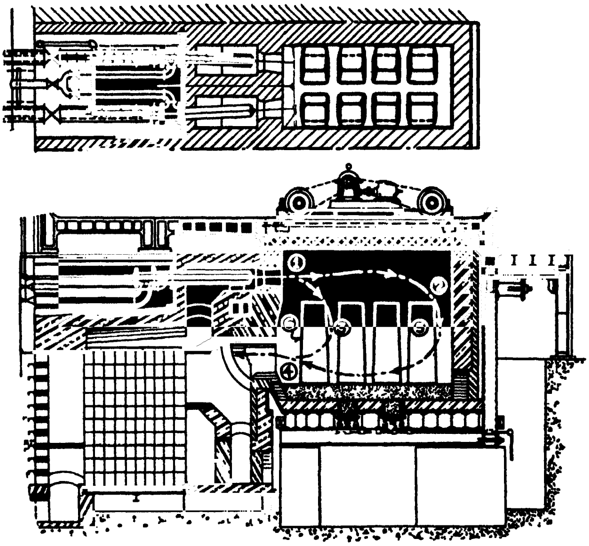


Fig. 335. One-way-fired soaking pit showing effect of initial velocity of flame.

(k) Circulation by Mechanical Means (Fans). In Chapter 3, the reasons were discussed why, in furnaces that carry less than 1400 F temperature, circulation by mechanical means is almost a necessity and why baffles are provided for guiding the circulation through the charge. Furnaces embodying the principle of mechanical circulation are illustrated by Figs. 10 and 60. Mechanical means circulate products of combustion, as well as protective atmospheres which are heated either by electric resistors or by radiant tubes. If the charge consists of a deep layer of small pieces, fans produce a very limited circulation only. For that reason, deep charges of small pieces are avoided in industrial furnaces.

For a given heating time, uniformity of temperature is increased; and for a given uniformity, heating time is shortened, if the direction of gas (or air) flow is periodically reversed. Reversal of flow is embodied in the furnace that is illustrated in Fig. 336. The fan is so designed that it is equally efficient in either direction.

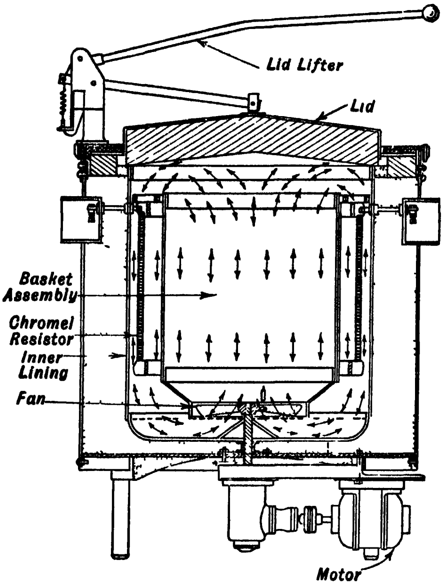


Fig. 336. Furnace with fan for producing circulation.

6. CIRCULATION OF GASES IN FURNACES OF VARIOUS TYPES

Flow of gases is a necessity in all furnaces that are heated by the combustion of fuel. In high-temperature furnaces, heat transfer by convection is negligibly small, but the products of combustion must flow and must be replaced by fresh, hot gases. In connection with Fig. 307, the statement was made that the gases should sweep over the charge when combustion has been almost completed in order to remove the cold layer on the charge.

A furnace that embodies this principle and looks somewhat like Fig. 307, but acts differently, is illustrated by Fig. 337. The furnace is intended to serve for temperatures below 1600 F. The stream of burning gases passes through the outlet flue, inducing a portion of the flue gases and returning them to the furnace where they are circulated around the stock, provided the latter is raised from the hearth a sufficient amount or is otherwise so arranged as to allow circulation. If the wall 3 were omitted, flow would not be nearly so regular as it is, because the stream would entrain gases almost exclusively from the top, and, besides, there would be no guarantee that the bottom layer of gas would be expelled through the outlet. In the design as shown, although there is an opening at 1 near the stack opening, no gas enters the flue from the furnace interior, because the flow is all in the other direction. Correct proportioning of the openings 1 and 4, of cross section of flue, and of length of path between burners 2 and opening 1 are necessary conditions for correct operation.

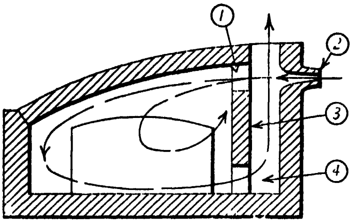


Fig. 337. Furnace designed for recirculation of products of combustion.

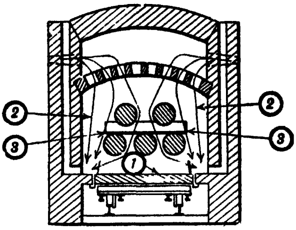


Fig. 338. Flow of gases in an overfired car-type furnace.

The gases circulate as described as long as the burner is wide open. If the burner is gradually turned down, a point is reached when the burning gases escape through the vent instead of entering the furnace.

In annealing furnaces, the temperature is usually too high for fans. Referring to Fig. 338, it will be noticed that the principal circulation is due almost entirely to pressure difference and location of outlet ports, while a slight secondary circulation arises from the fact that the walls are hotter than the material being heated. The effect of this secondary circulation is that a somewhat larger portion of the products of combustion follows the arrows 3 than might be expected from the restricted cross section, and that flow along the arrows 2 is slightly less than might be expected. It is well known among users of this type that it is extremely difficult to secure a uniform temperature, from top to bottom, at 1400 F, and that it is practically impossible to do so at 1200 F. The heating stock is always too cold immediately above the center of the hearth, that is to say at point 1.

For these reasons, overfired, or "perforated arch" furnaces, have become obsolete. The majority of such furnaces for annealing and heat treating at temperatures above 1400 F are underfired, with circulation obtained as illustrated in Fig. 308, where the injecting effect of modern burners causes gases from the heating chamber to be recirculated through the combustion chamber below the stock.

Cars have been built with flues in the hearth, which register with corresponding flues in the sidewalls (see Fig. 339). Temperature equalization is, of course, better than it is without the flues in the hearth, but it does not equal the uniformity that is obtained by forced circulation. Besides, the hearth has to be made thick, the car becomes heavy, and the hearth is more easily damaged during loading and unloading. In a few modern furnaces of this type auxiliary burners have been provided in the sidewalls immediately above the hearth and between the outlet ports. They produce some circulation and result in a more uniformly heated product.

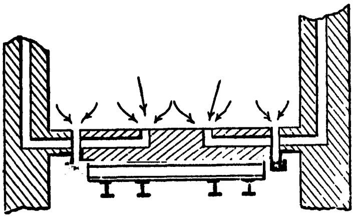


Fig. 339. Flues in hearth of furnace car.

In England, hearths of car-type furnaces have been built up of permeable bricks, through which an ejector sucks the products of combustion from the furnace. Circulation is excellent, but the hearth is weak.

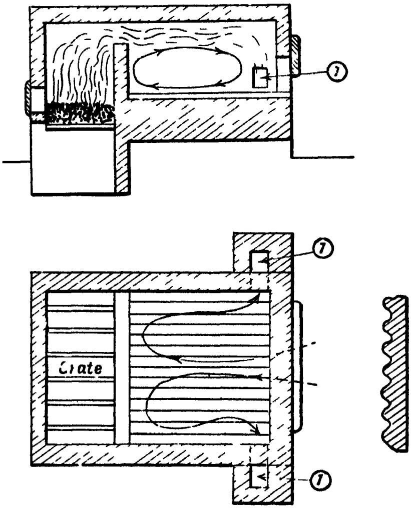


Fig. 340. Circulation in a sheet furnace.

An intensely interesting study of heat application and of circulation is furnished by the sheet-heating furnace, upon which for many years no improvements were made after it was brought to America by Welsh mill men. Although only a small number of furnaces of this type are now in use in the United States, it served its purpose so well that a study of the underlying principles is worth while. In the sheet furnace, a common form of which is diagrammatically shown in Fig. 340, it is extremely important that every trace of free oxygen be kept out, because scaling causes the sheets or packs of sheets to absorb furnace gases and to give trouble in rolling. On the other hand, there must be enough water vapor in the furnace atmosphere to produce a thin, even plastic coating of magnetic oxide, just thick enough to keep the sheets from sticking together when they are subjected to the excessively great pressure of the rolls. The furnace door is open during a large portion of the time; in some furnaces which heat light sheets or tin plate, it is open practically all the time. No air should flow in, however, and not enough furnace gas should come out to inconvenience the heater, who stands at the door working continuously, turning packs of sheets over, shifting them, or standing them up against the sidewalls.

With lump coal on the grate, alternate streaks of combustible matter and of air rise parallel in the space above the fire. Above the bridge wall they are mixed, and a secondary combustion sets in. Friction of the gases against the roof keeps up the mixing action, and a slowly moving, luminous flame rolls along the roof like an inverted river, with a fairly sharp demarcation against the rest of the furnace gases. The luminous flame heats the hearth and the sidewalls by radiation. The slow rolling motion of the luminous flame comes close to the motion that Groume-Grjimailo described as the result of his experiments with flow of non-mixing fluids of different density. There is, however, enough viscous drag to produce a very slow circulation, which is indicated by arrows in the illustration. The hot bridge wall helps this circulation along. The sheet furnace is a proof for the previously made statement that scaling is reduced to a minimum, if the stock lies in an unagitated atmosphere. The furnace temperature is low, never less than 1400 F and never more than 1600 F. For the sake of uniform rolling without curling, the packs must have the same temperature, top and bottom. This aim can be attained if the packs are laid on a bottom which has been previously heated by radiation in the empty furnace, and if the roof temperature is but little in excess of the final temperature which is desired for the sheets; but the rate of heating is, in that case, very slow. The sheets are laid on corrugated tile (see detail view to the right of Fig. 340), in order that the packs may be gripped easily, and also that gases may pass underneath. However, the quantity of gases passing under the sheets is very small indeed and contributes practically nothing to uniformity of heating. With the design of Fig. 340, circulation under the sheets may even be reversed by cold air which seeps in at the hearth level. While any air entering near the sides of the door is promptly taken into the flues, air near the center of the door can find its way in and circulate around, as indicated by the arrows in the plan view. To understand this situation, one must remember that the door is open to a width of 4 to 5 ft, and a height of 2 ft. If the furnace pressure is atmospheric at the level of the hearth, it must be in excess of atmospheric at the bottom of the raised door, and the flame and smoke will come out. As a matter of fact, smoke is intended to come out, and provision is made, by means of a hood, to carry it off. But the adjustment is so delicate that air does enter from time to time and causes the sheets to stick together in the mill.

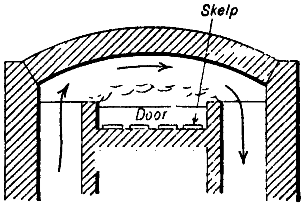
The old Welsh furnace men seem to have been fully alive to the situation, for their furnaces were built with ports in the hearth, and with flues under a section of the hearth. With that design, it is almost impossible for air to enter through the door, because the products of combustion form a screen or veil reaching all the way down to the hearth. This feature is excellent. The flues under the hearth are of doubtful value as far as keeping the hearth hot is concerned. American furnace builders discarded the flues under the hearth. In doing so, they also did away with the ports which opened into the underhearth flues. In conjunction with properly adjusted dampers and ashpit doors, these ports, which were located all along the door opening, kept all the air out of the furnace and considerably facilitated mill operations.

The introduction of the wide strip-mill caused furnaces such as shown in Fig. 340 to be abandoned, except in mills that produce small orders of sheets of different compositions. In those mills, natural gas, by product gas, and oil have replaced coal.

In some respects, the regenerative furnaces which are used in heating skelp, the strips from which welded pipe is made, are similar to the sheet furnaces. Gas and air (as in Fig. 341) come up on one side through separate and alternate ducts and combine in a luminous flame which rolls along the roof and passes down on the other side to the regenerators. Heating is almost exclusively by radiation and not by circulation of the gases between the pieces of the charge. In view of the thinness of the skelp, this method is very satisfactory, as far as uniformity of temperature in the steel is concerned; besides, very little scaling occurs if air is kept out of the furnace.

Skelp-heating furnaces are quite long, in the direction at right angles to the section shown in Fig. 341. The products of combustion pass down through a regenerator (or two regenerators, if producer gas serves as fuel) on one side, while air comes up through hot checkerwork on the other side. It takes a skillful designer so to proportion the air approach and the flues under the regenerators that the upward flow of air is uniform along the furnace. Since that desirable goal is seldom attained, adjustable dampers are provided on top of the ports. The furnace temperature lies between 2600 and 2800 F (the edges of the skelp must be dripping hot); for that reason, the dampers are made of high-alumina tiles or of silicon carbide.

Fig. 341. Circulation in a regenerative furnace for heating skelp.



In continuous butt-welding mills, the furnace illustrated by Fig. 341 is replaced by the continuous skelp-heating furnace illustrated by Fig. 74.

In continuous furnaces for heating billets, the latter may be pushed down on an inclined hearth, up an inclined hearth, or horizontally. The theory has been advanced that the only correct method is to push the billets upward, because cold gases were supposed to stay at the bottom (which is the discharge end of the furnace in the usual arrangement shown in Fig. 342), while the hot gases rise to the top (which is the charging end of the furnace in Fig. 342). That theory, which, by the way, was derived from observation of experiments on the flow of colored kerosene and water through furnace models, is wrong in this case as well as in many others. Several factors combine to make it so. Even if the flame (and flame is visible in most of these furnaces) does not touch the stock at the hot end, viscous drag sets up a secondary circulation (as indicated at 1, Fig. 342), and this circulation is aided by the arched shape of the roof. Furthermore, radiation is so very active at the hot end that the stock would be heated even without the beforementioned secondary circulation. (In 1911, when the tests using non-miscible liquids of different color in furnace models were published, the fact that clear gases radiate heat was practically unknown.) Last, but not least, the upward flow of the products in the furnace produces a gentle chimney effect, which, in conjunction with a reduction of furnace cross section, acts to maintain atmospheric pressure throughout the length of the furnace, at least for normal heating conditions. Products of combustion are not discharged into the open through the doors, and cold air does not enter.

In a few furnaces in the United States, billets were pushed upward to a high point from which they slid down to a mill table. This arrangement was not chosen on account of heat distribution or circulation in the furnace. It was chosen because it saved a crane lift and labor.

Incidentally, Fig. 342 shows the necessity for correct circulation of furnace gases at comparatively low furnace temperatures. When the furnace has been in operation for some time, the ends of the pusher bars burn off, and the hearth becomes covered with billets to the very end, as indicated at 2. In that case, the flue gases can escape only through the cracks between the billets, and at the sides, at points 3 and 5. A cold triangle 4 is the result of this faulty circulation.

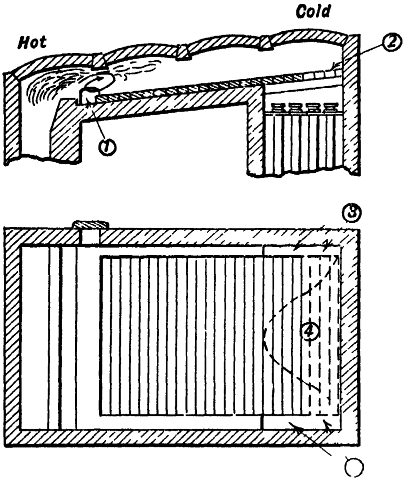


Fig. 342. Flow of gases in a continuous billet-heating furnace.

The above mentioned nicety of maintaining neither excess pressure nor draft (deficiency of pressure) along the hearth of Fig. 342 exists for the heating rate contemplated by the originators of the design. That rate was approximately 65 lb of steel per (sq ft of effective hearth area, hr). Speeding the mill, or using two stands in the mill has led to changes in the furnace, whereby the rate has been pushed up to more than 130 lb per (sq ft, hr). Pushing approximately twice as much fuel and air through the furnace as intended by the designers, makes flames escape around the trouble-doors, most of which were bricked up. The recuperator reduces the draft (see the section on regenerative furnaces), and had to be taken out. It was replaced by burners under the skids, where the steel enters.

Additional examples on gas circulation are given in Volume II, especially in the chapter on control of furnace atmosphere.

The above discussed examples add up to the rule. "Keep furnace gases away from the charge in high-temperature furnaces, but bathe the charge in the furnace gases of low-temperature furnaces."

APPENDICES

appendix 1

Distribution of Transient Temperatures in Solid Bodies

Industrial furnaces can be designed, constructed, and operated on the basis of the data that are contained in the eight main chapters of this book. For those who are not satisfied with designing by rules of experience, several methods are available for investigating the time-temperature-space relations that exist when a solid body is being heated or cooled. Among these methods are:

(*a*) Charts and tables for plates, cylinders, and spheres.
(*b*) Graphical methods for plates, cylinders, and spheres.
(*c*) Numerical step-by-step calculations.
(*d*) Experimental methods.

(a) Tables and Charts. By the use of Fourier functions and of Bessel functions, numerical values were computed for temperature as a function of time and of location in the solid. These values were either put into tabular form or converted into graphical charts.

The analytical method necessitated the introduction of several limitations and simplifications: namely, (*a*) the solid is homogeneous; (*b*) at the beginning of the heating or cooling period, the temperature is uniform in all parts of the solid; (*c*) the coefficient of heat transfer is constant; (*d*) the ambient temperature is constant; (*e*) the thermal conductivity of the solid is constant; (*f*) the specific heat (thermal capacity) of the solid is constant. In spite of these limitations and unrealities, the tables and charts render good service, if judgment is exercised in selecting representative mean values for the variables.

In order to make the tables and charts applicable to any actual case, they were computed or plotted for dimensionless coordinates such as, for instance,

$$\frac{\text{Diffusivity} \times \text{Time}}{\text{Radius squared}} = \frac{\text{ft}^2}{\text{hr}} \times \frac{\text{hr}}{\text{ft}^2}$$

$$\frac{\text{Heat-transfer coefficient} \times \text{Radius}}{\text{Thermal conductivity}} = \frac{\text{Btu}}{\text{ft}^2\text{,hr,F}} \times \text{ft} \times \frac{\text{ft,hr,F}}{\text{Btu}}$$

The reciprocal of these coordinates is, of course, also dimensionless, and so is a ratio of temperatures. Some mathematicians (for instance Bachmann) based their calculations on other dimensionless coordinates.

The results of calculations are obtained in numerical form. They may be tabulated in that form for practical use. Several such tabulations have been published. One of the best known sets of tables was computed by T. F. Russell and appeared in the "First Report of the Alloy Steel Research Committee, Special Report 14, of the British Iron & Steel Institute, 1936." In the United States, Russell's tables were

TABLE 23

Plate. $HL = 1$

	Value of U when x/L is						
τ	0.0	0.5	0.6	0.7	0.8	0.9	1.0
0.05	0.99994	0.98654	0.97338	0.94963	0.91333	0.86080	0.79062
0.10	0.99332	0.95073	0.92539	0.89106	0.84643	0.79074	0.72384
0.20	0.94864	0.87950	0.84637	0.80627	0.75904	0.70481	0.64365
0.30	0.89202	0.81550	0.78183	0.74215	0.69663	0.64552	0.58912
0.40	0.83119	0.75696	0.72478	0.68718	0.64439	0.59670	0.54444
0.50	0.77455	0.70285	0.67269	0.63755	0.59765	0.55328	0.50479
0.60	0.71793	0.65269	0.62459	0.59188	0.55478	0.51356	0.46853
0.70	0.66681	0.60612	0.57999	0.54961	0.51513	0.47685	0.43500
0.80	0.61929	0.56290	0.53863	0.51039	0.47837	0.44282	0.40398
1.0	0.53411	0.48547	0.46454	0.44018	0.41257	0.38190	0.34841
1.5	0.36895	0.33535	0.32090	0.30407	0.28499	0.26380	0.24068
2.0	0.25487	0.23166	0.22167	0.21005	0.19687	0.18223	0.16625
3.0	0.12162	0.11054	0.10578	0.10023	0.09394	0.08696	0.07933
4.0	0.05803	0.05275	0.05047	0.04783	0.04483	0.04150	0.03786
5.0	0.02769	0.02517	0.02408	0.02282	0.02139	0.01980	0.01806
6.0	0.01321	...	...	...	...	...	0.00862
7.0	0.00630	...	...	...	...	...	0.00411
8.0	0.00301	...	...	...	. .	...	0.00196
9.0	0.00144	...	...	...	...	...	0.00094

reprinted in "The Flow of Heat in Metals," by J. B. Austin, American Society for Metals, Cleveland, Ohio, 1942. The complete set consists of 19 tables for plates and cylinders. One of the tables is reproduced here as Table 23, in order to explain the arrangement and the symbols. The heading $HL = 1$ signifies that the table was calculated for (heat-transfer coefficient/conductivity) × ½ thickness = 1. The Greek letter τ over the left-hand column stands for (time × diffusivity)/(½ thickness)^2. The fraction at the head of each vertical column equals the ratio (distance from center)/(½ thickness); U is the ratio: (not yet completed temperature change)/(maximum possible temperature change). In Russell's tables, the values of HL (using Russell's notation) are so far apart that correct interpolation is extremely difficult.

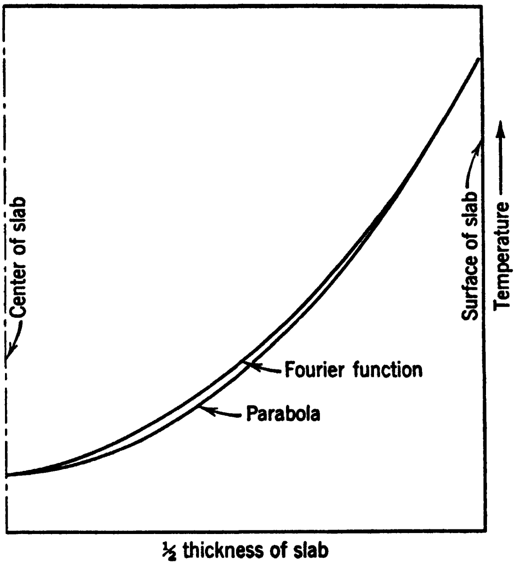


Fig. 343. Fourier curve and parabola.

After a fraction of the total heating or cooling time has passed, the temperature curve in the solid assumes a shape that approaches a parabola so closely that the parabola may be substituted for the Fourier curve. The admissibility of such a substitution was tested by plotting (in Fig. 343) a temperature curve for rapid heating from Table 23 and by entering a parabolic arc on the same illustration. Even in this extreme case, the approximation is close; and, since the Fourier curve is only an approximation to the curve of true temperature distribution, the substitution is justified.

Most engineers prefer graphical charts to tables, because trends are more easily observed. This fact is evidenced by the large number of different charts that have been plotted from the results of numerical calculations of Fourier functions and Bessel functions. For practical purposes, the charts originated by Hottel and Heisler are very suitable (see "Temperature Charts for Induction and Constant-Temperature Heating," by M. P. Heisler, *Transactions of the American Society of Mechanical Engineers*, April 1947). Heisler's charts are here reproduced (by permission) in Figs. 344 to 349. In Figs. 344, 345, and 346 the ordinates are not-yet-completed temperature-change ratios at the center of the solid; the abscissae represent the fraction $\frac{\text{Diffusivity} \times \text{Time}}{(\frac{1}{2}\text{ Thickness})^2}$. The parameters m represent the fraction

$$\frac{\text{Conductivity}}{\frac{1}{2}\text{ Thickness} \times \text{Heat-transfer coefficient}}$$

Figs. 344, 345, and 346 show the temperature ratio at the center only. If it is attempted to plot on the same chart, temperature ratios for points that are located at various distances from the center, either the chart becomes a maze of lines which cannot be disentangled, or else so few values can be shown that interpolation becomes extremely inaccurate. The dilemma is avoided by constructing either a separate set of charts for the surface (Hottel) or a set of auxiliary curve sheets which contain modifying factors for points located at various distances from the center. The latter step was taken by Heisler. Figures 347 to 349 are his auxiliary charts.

The use of the charts (Figs. 344 to 349) is best illustrated by an example. Let an ingot of very soft steel, 36 in. in diameter at 100 F, be placed in a furnace at 2300 F.* How long must it stay in the furnace for the center of the ingot to reach 2180 F, and what will be the temperature difference between surface and center at that time?

The first step is to select representative mean values for heat-transfer coefficient k, conductivity C, and also for density and specific heat, because they enter into diffusivity j. By reference to tables and charts in former chapters, the following values are found: $C = 23$ Btu per (ft, hr, F); $j = \frac{\text{Conductivity}}{\text{Spec. heat} \times \text{Density}} = 0.35$ sq ft per hr; $m = C/(rk) = \frac{23}{1.5 \times 20} = 0.77 =$ Dimensionless parameter. Not-yet-completed temperature-difference ratio $= \frac{2300\text{ F} - 2180\text{ F}}{2300\text{ F} - 100\text{ F}} = 0.005$ (dimensionless ordinate). From Fig. 345, the abscissa which corresponds to the intersection of ordinate 0.055 and parameter 0.77 is $1.65 = jt/r^2$. From this equation, $t = 1.65 \times 2.25/0.35 = 11.6$ hr.

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* This method of heating is not recommended for high-carbon steels and for tender alloy steels (see page 69).

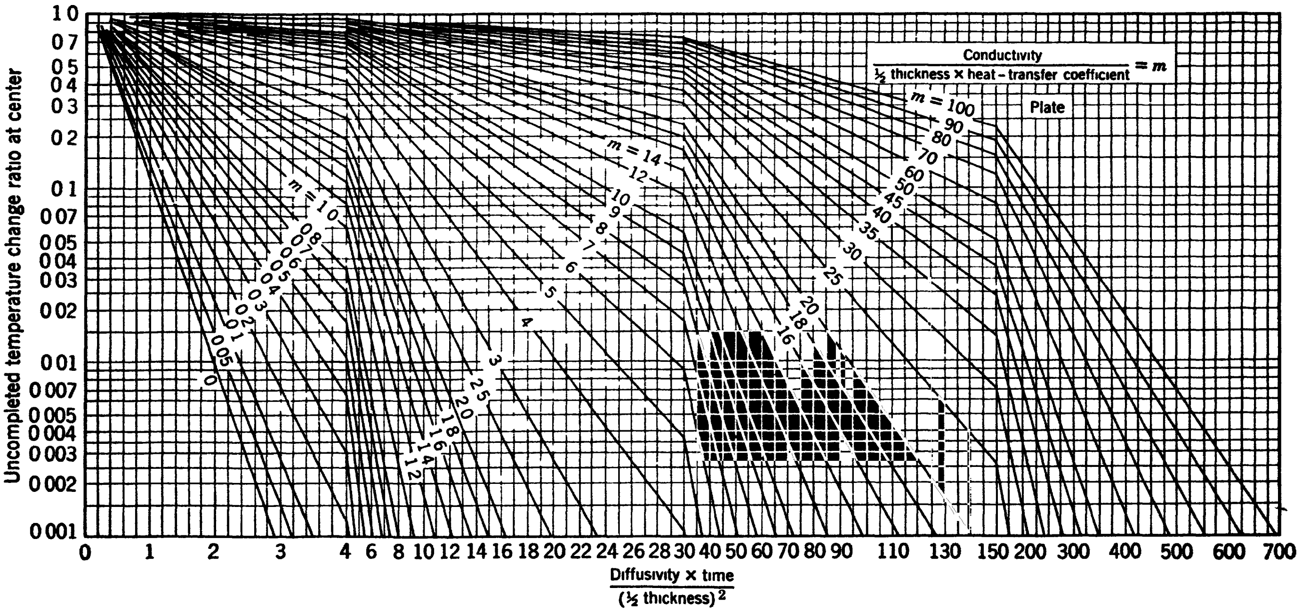

Fig. 344. Temperature-time-space relation for center of large plate (Courtesy of American Society of Mechanical Engineers)

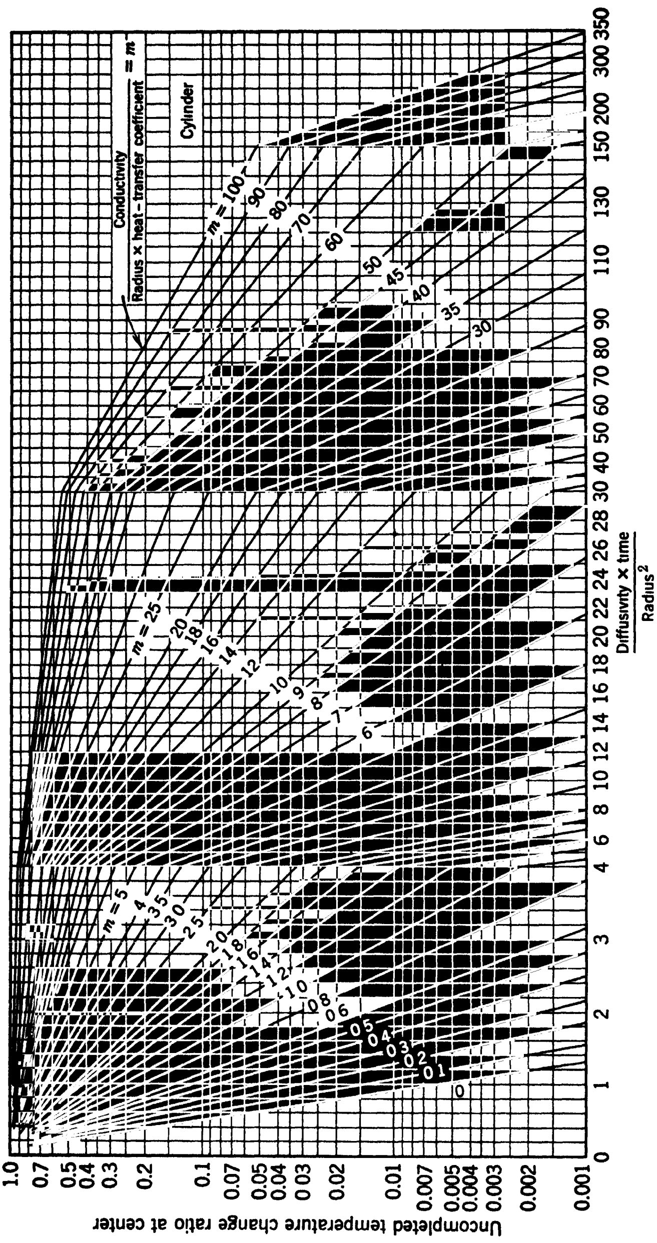

Fig. 345. Temperature-time-space relation for axis of long cylinder. (Courtesy of American Society of Mechanical Engineers.)

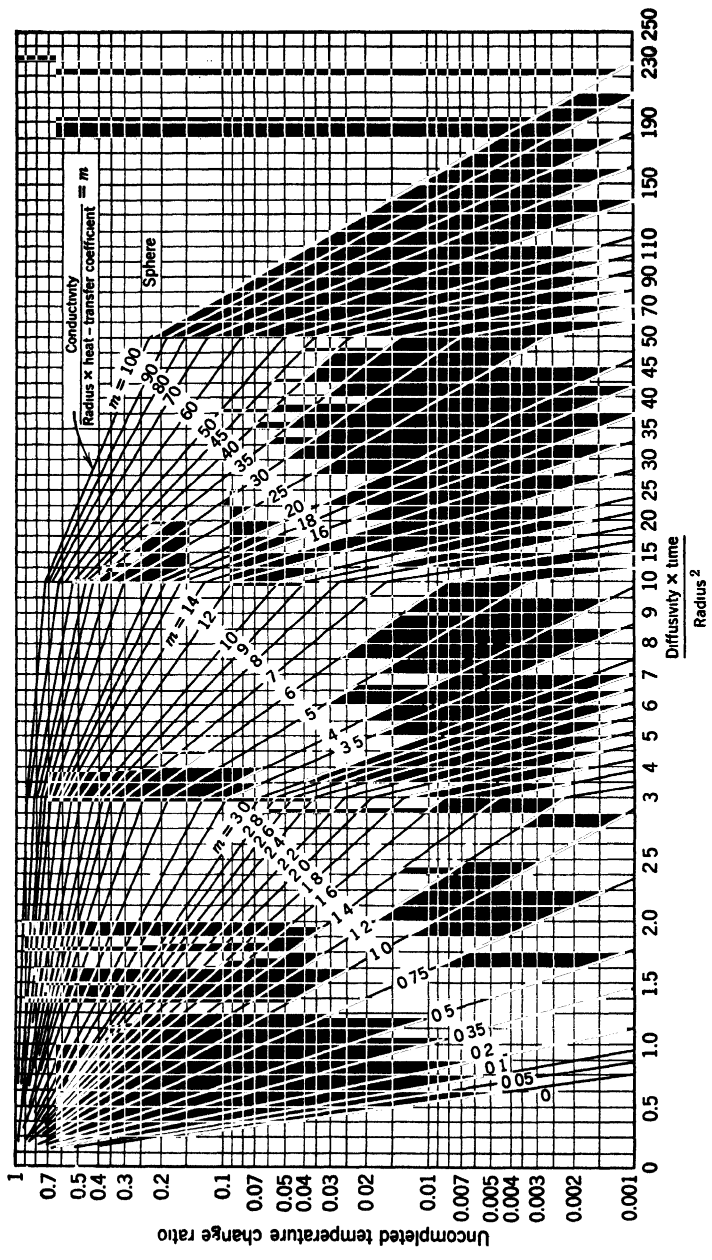

Fig. 346. Temperature-time-space relation for center of a sphere. (Courtesy of American Society of Mechanical Engineers.)

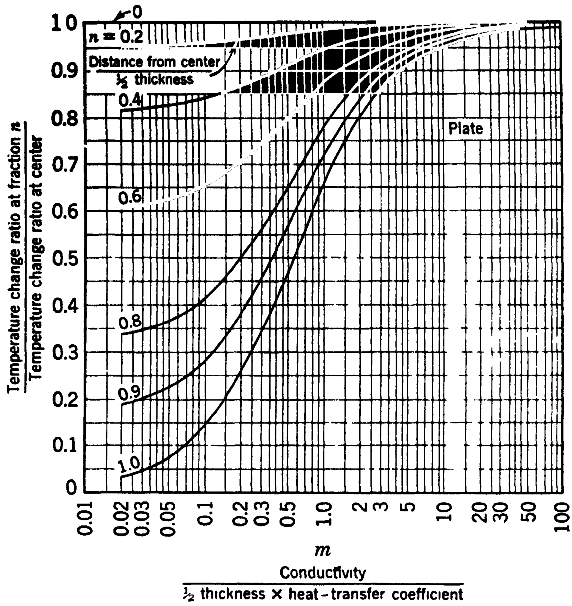

Fig. 347. Extension of Fig. 344 to away-from-center locations. (Courtesy of American Society of Mechanical Engineers.)

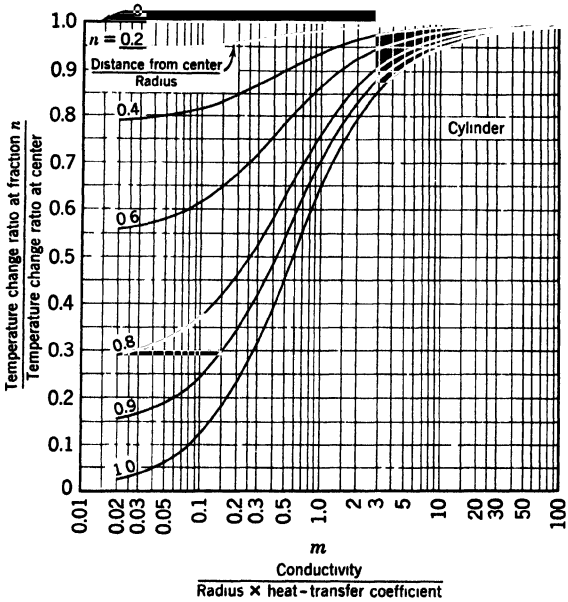

Fig. 348. Extension of Fig. 345 to away-from-center locations. (Courtesy of American Society of Mechanical Engineers.)

The surface temperature is found from Fig. 348. In this illustration, *n* is the ratio of radius at any point to radius at the surface. In the present example, $n = 1$. Then, for $n = 1$ and $m = 0.77$, the temperature-difference ratio is 0.56. Hence, the not-yet-completed temperature difference (at the surface) equals 0.56 × 120 F = 67 F. The surface temperature is 2300 F − 67 F = 2233 F. The difference of temperature between surface and center equals 2233 F − 2180 F = 53 F. The same answer is, of course, obtained by the expression (1 − 0.56) × 120 = 53 F.

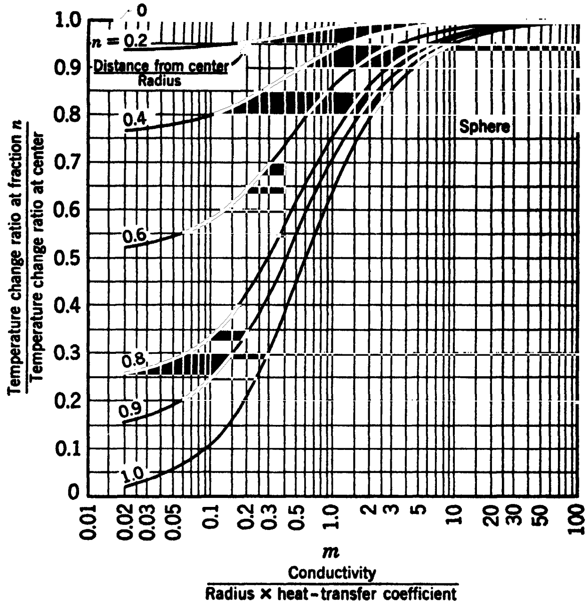

Fig. 349. Extension of Fig. 346 to away-from-center locations. (Courtesy of American Society of Mechanical Engineers.)

If the flame of 2300 F is luminous, *k* may rise to 40 Btu per (sq ft, hr, F). In that case, $m = 0.39$, and the abscissa becomes 1.1; then $t = 1.1 \times 2.25/0.35 = 5.8$ hr, which is approximately equal to 10 min per in. diameter. The surface temperature is 2300 F − 0.38 × 120 F = 2254 F.

If heat flows into a rectangular body from different directions, the charts (Figs. 344 and 347) can still be used. In the May 1936 number of *Industrial and Engineering Chemistry*, A. B. Newman proved that the actual not-yet-completed temperature difference can be obtained by multiplication of the temperature-change ratios in the three principal directions. If the temperature of a point (defined by the coordinates *a*, *b*, *c*, in Fig. 350) at a given time is to be found from Figs. 344 and 347, the following method is used: From Fig. 344, find the not-yet-completed temperature-change ratio for the center of the block in the principal directions; modify them from Fig. 347 in the three principal directions for $m = a/A, b/B$, and c/C. Form the product of the three thus obtained temperature-change ratios. The product equals the desired, not-yet completed temperature-change ratio for the point under consideration. Evidently, the product of three proper fractions is smaller than any of the individual fractions. On page 547 of Newman's paper an example is given for heating a brick which receives heat from all sides.

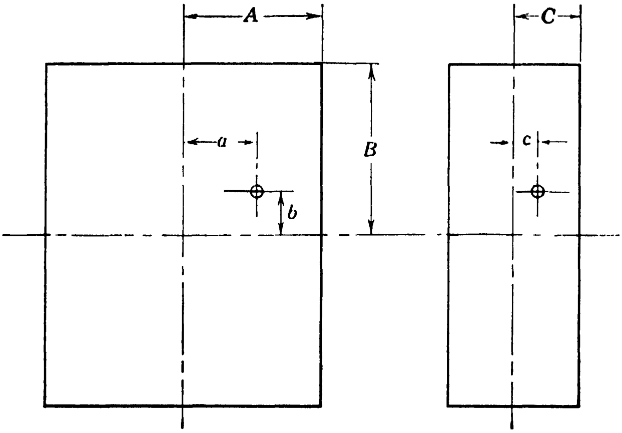

Fig. 350. Temperature-time-space relation for heat flow in three directions.

(b) Graphical (Schmidt's) Method. Although Heisler's charts are very convenient in their use, they (as well as similar charts) are subject to severe limitations, the most objectionable of which is the requirement of uniform temperature at the beginning of the heating or cooling process. A graphical method that was developed by E. Schmidt* for the heating or cooling of flat plates or slabs is free from this limitation.

Considering Fig. 351, let ds be a small but finite element or portion of the plate thickness and dt be a finite interval of time; let T_1, T_2, etc., be temperatures; C = conductivity; q = density; c_m = specific heat (thermal capacity)—all referring to the material of the plate.

Then $(T_4 - T_3) \times C \dfrac{dt}{ds}$ = heat transmitted in time dt through unit area of element 3, 4.

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* "Beiträge zur technischen Mechanik und technischen Physik," by students of Aug. Föppl, page 179 (Springer, Berlin, 1924).

$(T_3 - T_2) \times C \dfrac{dt}{ds}$ = heat transmitted in time dt through unit area of element 2, 3.

Difference $(T_4 - 2T_3 + T_2)C\dfrac{dt}{ds}$ = heat stored in element ds located symmetrically on both sides of plane 3, which equals $(T_3' - T_3) \times q \cdot c_m \cdot (ds)$; hence the equation

$$(T_3' - T_3) = \frac{C}{qc_m}\frac{(dt)}{(ds)^2}(T_4 - 2T_3 + T_2) \tag{41}$$

in which T_3' and T_3 are temperatures in plane 3 at the end and at the beginning of the time interval dt.

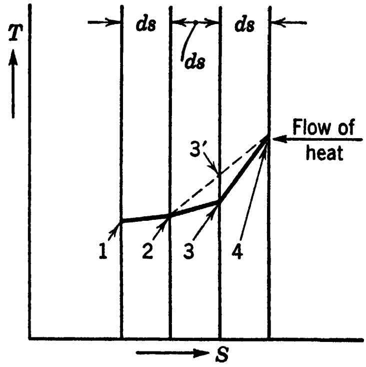

Fig. 351. Temperature relations in the heating of a plate.

If the terminals of temperatures 2 and 4 be joined by a straight line, then the distance 3-3′ equals $\frac{1}{2}(T_4 - 2T_3 + T_2)$, because the temperature 3′ equals $\frac{1}{2}(T_4 + T_2)$. On multiplying 3-3′ by $\dfrac{2C}{qc_m} \cdot \dfrac{(dt)}{(ds)^2}$, the product obtained, according to equation 41, is the temperature rise in the plane 3 in time dt. Time interval dt and space increment ds can be selected arbitrarily, but must be small enough to produce accurate results. The graphical computation is very simple, if $\dfrac{2C}{qc_m}\dfrac{(dt)}{(ds)^2}$ is made equal to unity, or $\dfrac{dt}{(ds)^2} = \dfrac{1}{2}\dfrac{qc_m}{C}$, because, in that case, 3′ furnishes the terminal of the temperature at the end of time interval dt.

In technical problems, heat transfer at the surface of the plate must be taken into consideration. It is customary to assume that heat transfer is proportional to difference of temperatures of surface of wall T_s and of surrounding medium T_m, or, in other words, that k, the coefficient of heat transfer between the surface of the plate and the medium, is independent of the temperature level. Then, referring to Fig. 352,

$$k(T_m - T_s) = C\frac{dt}{ds} \qquad (42)$$

where the left-hand member of the equation equals the heat transmitted from the medium to unit area of the plate in unit time, and the right-hand member equals the heat conducted through unit area of the surface of the plate in unit time. The coefficient k depends on many circumstances. Values can be found on pages 61 to 63.

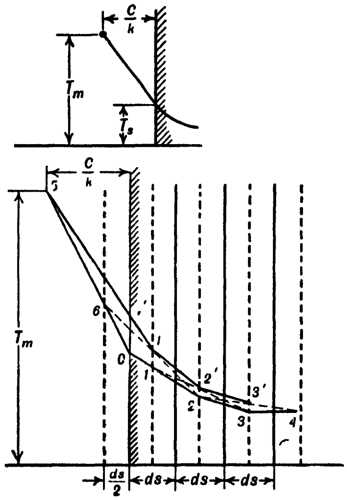

Fig. 352. Geometrical relations applying to flow of heat in a plate.

Equation 42 can be interpreted geometrically. The tangent to the temperature curve at the surface of the plate passes through a point located at elevation T_m, and C/k distant from the wall (see Fig. 352). If T_m or k varies with time, such variation can be taken care of by the graphical computation, whereas that cannot be done in an analytical calculation. The graphical computation proceeds as follows:

Let 0, 1, 2, 3, 4, in Fig. 352 be the temperature distribution in the beginning of time interval dt. Let 5 be the point T (determined by T_m and C/k) toward which the surface tangents to the temperature curve point. The temperatures 2′, 3′, at the end of interval dt are found as before. To determine 1′, join points 0 (at surface) and 5; find intersection 6 with a straight line $\frac{1}{2}ds$ distant from the surface of the plate. Join 6 and 2 and find intersection 1′ with vertical line through 1; then 1′ is the desired temperature of point 1 at the end of time dt. Join 1′ and 5, and find intersection 0′ with the surface. Point 0′ determines the temperature of the surface at the end of time dt.

The broken line 0′, 1′, 2′, 3′, etc., represents the temperature at end of time *dt*. It forms the base line for the next time interval *dt*. Point 5 is moved to a new position, if temperature T_m or coefficient *k* changes.

An example will illustrate the use of this method. The heating of a brick wall 4½ in. thick will be studied. Although, for the sake of accuracy, the heat-transfer coefficient should be varied with the wall temperature, it was kept constant in the example for purpose of comparison with the analytical method of solution. The wall was divided into eight sections, through the center of each of which a construction line was drawn. Values of *ds*, *q*, c_m, *C*, *k*, and the calculation for *dt* are given in Fig. 353. For greater clearness, the method of starting the construction in Fig. 353 is shown step by step in Fig. 354.

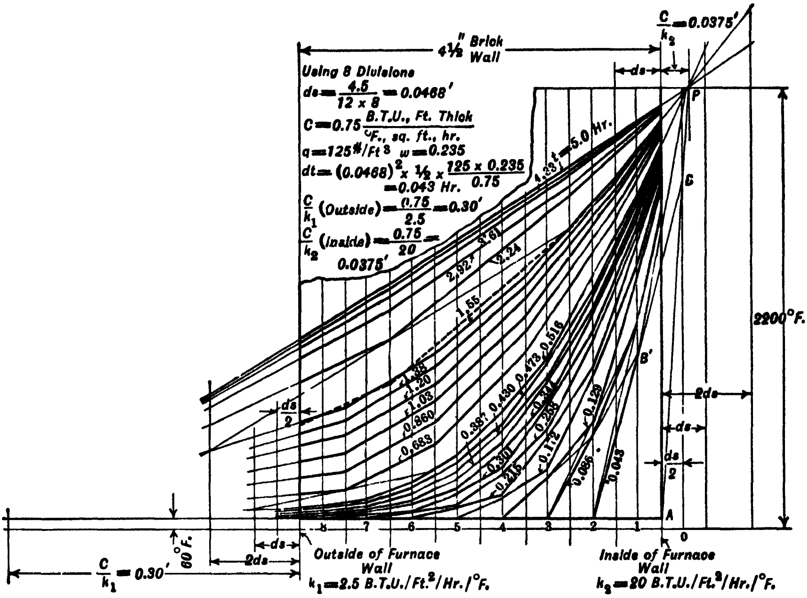

Fig. 353. Example of graphical method of finding heating time and temperature distribution in a furnace wall. Furnace temperature = 2200 F; temperature of outside air = 60 F. Wall is of firebrick 4½ in. thick. Specific heat = 0.235.

If one side of the wall is suddenly exposed to a temperature of 2200 F, the tangent to the temperature curve at point *A* must point to pole *P*; hence the intersection *B* between *AP* and the vertical line through point 0 (at distance ½*ds* beyond the surface) is first located (first step, Fig. 354), and this point is the terminal of the first construction line 2-*B* in the second step. Line 2-*B′* is part of the first temperature polygon; the latter is completed by drawing a line from

B′ to pole *P*. The line 2-*B′*-*D* then represents the temperature in the wall after an interval of 0.043 hr after the start of heating. In the third step, construction line 2-*E* is drawn, and its intersection with the vertical through 1 locates point *F*. Construction line 3-*B′* intersects the vertical through point 2, locating *G*. In the fourth step, lines are drawn connecting 3, *G*, and *F*, and from *F* a line is drawn to pole *P*. The temperature distribution after 2 × 0.043 = 0.086 hr is then represented by 3-*G*-*F*-*J*.

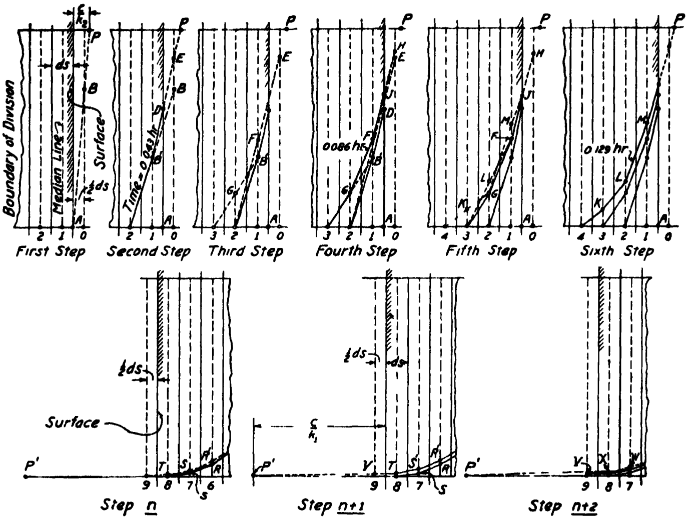

Fig. 354. Illustrating steps in construction of Schmidt diagram.

The fifth, sixth, and following steps proceed in the same manner, the right-hand end of each temperature polygon being drawn from the point of intersection with the vertical through 1, as *F* or *M*, directly toward the pole, and the left-hand part of each temperature polygon ending on the base line, as at points 3 and 4. After this end of the curve has reached point 8, as in step *n*, for the next curve points *R′* and *S′* are determined as usual, while *T* is determined by drawing a construction line from *S* to point 9 on the base line at a distance $\frac{1}{2}ds$ to the left of the boundary line (plane of the surface of the wall). In step $(n + 1)$, the left-hand end of the temperature polygon is drawn from *T* directly toward pole *P′*, and in all following steps the left-hand end of the polygon points to this pole. Each temperature polygon must be completed before the next is started. The time from the start of heating, corresponding to the *n*th temperature curve drawn, is of course *n* times the interval *dt* between successive curves.

Successive temperature curves come closer and closer together. Before confusion occurs, it is advisable to jump every other lamina. The time interval is thereby multiplied by four. In doing so, one should redraw the last preceding temperature polygon in such a way that it consists of straight lines extending through two laminae, forming a polygon of the same area as the one with twice the number of sides (see Fig. 353, broken-line polygon for 1.55 hr). It is also advisable to erase the original vertical division and construction lines above this polygon, and draw in only the center lines of the new doubled laminae, in order to avoid confusion of lines. (Even at the start of the construction, the boundary lines of the laminae may well be omitted, leaving only the median lines.)

If, on doubling the laminae, the new vertical line *OB* located at a distance beyond the surface equal to ½ the width of the new lamina is found to be farther from the surface than pole *P*, then the construction using points of intersection with this line, as *B*, is to be omitted, and pole *P* is to be used as the end point of all construction lines used in determining points on the median line of the first lamina.

Toward the last, when the polygons flatten out, the width of the laminae may again be doubled and the time quadrupled. Extreme flattening is avoided by making the chart quite tall, that is to say, long in the direction of the temperature scale.

A comparison of the result of the graphical determination with the result of analytical calculations using Fourier functions shows a close agreement. The results of other examples have shown that, when the wall thickness is divided into four or more laminae, the results of the graphical method are quite accurate, but, when less than four divisions are used, the discrepancy becomes considerable.

For the case in which a plate or slab is heated equally from both sides, only the half-thickness need be considered. The sections of the polygons crossing the middle plane of the plate are then drawn horizontally in every case, joining the median lines of the two laminae immediately on either side of the center line of the plate. Not less than four divisions should be used in the half-thickness. Some engineers find it difficult to carry the construction lines properly across the center. For their benefit Fig. 355 is offered.

In his paper on the graphical method, Schmidt brought out several interesting facts. The temperature of the slab need not be uniform at the beginning of the heating or cooling period. Variations of the temperature T_m of the surrounding medium, as well as changes in the heat-transfer coefficient *k*, are easily introduced into the construction. Schmidt also explained and illustrated the changes that must be made in the diagram if the slab or wall is composite (two or more slabs of different materials in contact). Unsteady flow through composite walls was discussed in detail and a modification of Schmidt's diagram was given by T. C. Patton in *Industrial and Engineering Chemistry,* November 1944.

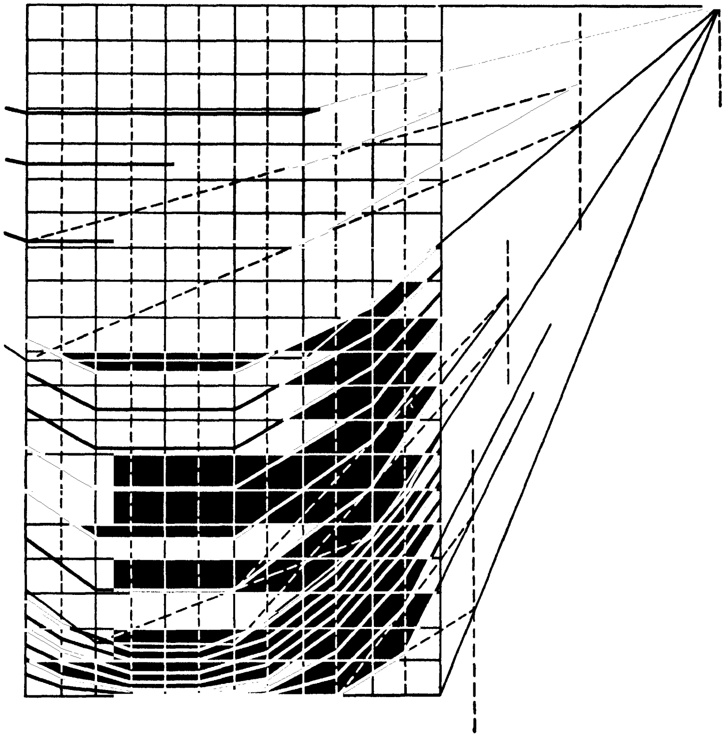

Fig. 355. Schmidt diagram for symmetrical bilateral flow of heat into a plate.

Graphical Method for Finding Temperature Distribution in Cylinders and in Spheres. Schmidt (*loc. cit.*) stated that a difference equation can be derived for circular cross sections but that the resulting "Rekursionsformel" (reiterative equation) is not simple because it contains a term that depends on the curvature.

In spite of the lack of simplicity, several mathematicians have derived graphical methods for long cylinders and for spheres. In *Chaleur et Industrie,* 1928, Nessi and Nissole published such a method. P. S. Austen, of Glen Ridge, N. J., independently conceived the method (private communication to the author). T. C. Patton (*loc. cit.*) applied the same method to thick cylindrical shells

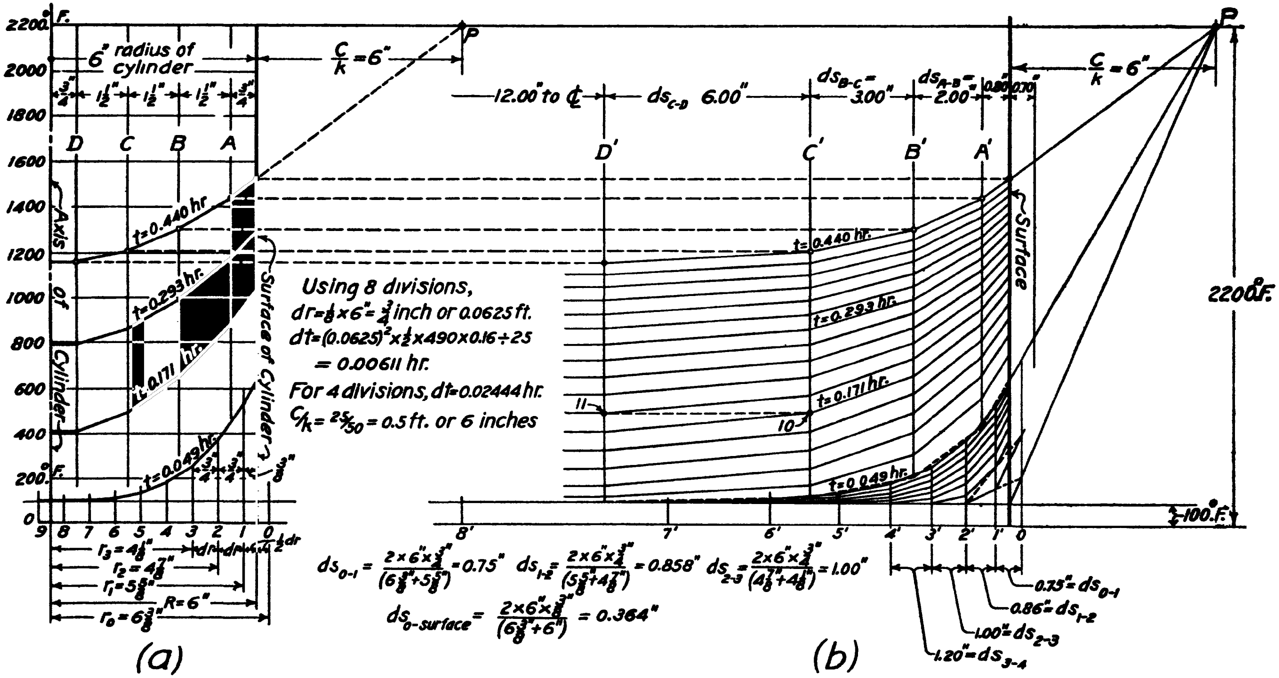

Fig. 356. Graphical method of finding heating time and temperature distribution in a solid cylinder. Furnace temperature = 2200 F; initial temperature of cylinder = 100 F. Cylinder is of mild steel 12 in. in diameter. Average conductivity = 25 Btu per (ft, hr, F). Specific heat = 0.16 Btu per (lb, F). Surface heat-transfer coefficient = 50 Btu per (sq ft, hr, F).

The method consists of modifying the spaces or laminae by making their widths inversely proportional to the radius. Thus, in Fig. 356, the left-hand diagram *a* representing the half-thickness of the solid cylinder, divided into spaces of equal width dr with median lines 1, 2, 3, 4, etc., is replaced by the modified diagram *b* at the right having the same number of divisions but with the distances between the new median lines 1′, 2′, 3′, 4′, etc., of the spaces increasing from the surface to the center. If R is the radius to the outside of the cylinder, r_1, r_2, r_3, etc., the radii to median lines 1, 2, 3, etc., and r_0 the radius to the construction line at a distance $\frac{1}{2}dr$ beyond the surface of the cylinder, then the distances between the corresponding median lines in the modified diagram Fig. 356*b* are determined by the equations:

$$\left.\begin{aligned} ds_{1-2} &= \frac{R \cdot dr}{\frac{1}{2}(r_1 + r_2)} \\ ds_{2-3} &= \frac{R \cdot dr}{\frac{1}{2}(r_2 + r_3)} \\ ds_{0-1} &= \frac{R \cdot dr}{\frac{1}{2}(r_0 + r_1)} \end{aligned}\right\} \qquad (43)$$

The pole distance is exactly the same as for flat plates, or C/k. The time interval is also the same, or $dt = \frac{1}{2}(dr)^2 \times$ density $\times$ specific heat/conductivity.

For the sake of compactness, the center line of the cylinder and the median line on the other side of it have not been shown in the right-hand part *b* of the illustration. In fact, the error is very small (when the number of divisions is not less than four) if these two lines are omitted altogether from the construction, and points such as 11 on the median nearest the center line, as D', are obtained by projecting horizontally to the left from points such as 10 on the adjacent median C', to intersect line D' (see broken line 10-11). When changing scales, as from the narrow spaces marked with numerals 1, 2, 3, etc., to the wider spaces marked A, B, C, D, if the number of divisions of the actual radius is halved, the time interval is quadrupled, just as for flat plates. The positions of the new median lines for the wider spaces are again figured from equation 43. The last preceding temperature polygon is redrawn as explained on page 449 (see broken line for $t = 0.049$ in Fig. 356*b*). The accuracy of the method is good. Even if only four divisions are used in the half-thickness or radius of the cylinder, the error (with careful graphical work) seldom exceeds 2 per cent of the result obtained by the long-drawn-out analytical method using Bessel functions. By projecting the points on the new median lines in part *b* of the illustration, horizontally across to the old median lines on part *a*, the true form of the temperature-space curve can be obtained.

The Schmidt graphical method was extended to spheres by Keller (*Industrial Heating*, October 1939) and by Patton (*loc. cit.*). The graphical method for spheres is very similar to that used for cylinders, but, whereas the width of the spaces in the cylinder construction is distorted inversely proportional to the mean radius, it is, for the sphere, distorted inversely proportional to the square of the mean radius.

(c) Numerical Calculation. The temperature distribution in solids, either for the steady state or for transient conditions, may be calculated numerically by processes that require much time. Many papers and books have been published on numerical methods. One of the most complete books is: G. M. Dusenberre, *Numerical Analysis of Heat Flow*, McGraw-Hill (1949). Inquiry about a new edition revealed that the first edition is out of print and that no later edition is available. From these facts the authors of the present book judge that the very slow and tedious method of numerical calculation has not found favor with furnace engineers.

(d) Experimental Methods. The flow of heat in solids can be simulated by hydraulic analogies and by electrical analogies. The latter method is convenient, because it furnishes results in a short time and permits accurate measurements of some variable quantities. It was originated by C. L. Beuken (*Eonomish Technish Tijdschrift*, Mastricht, Holland, 1937). The continuous solid is replaced by a number of stations with wires (resistances) between them. Heat storage is simulated by capacitors (condensers). Very feeble currents flow in the apparatus. For applicability of the resistance-capacity arrangement see *Transactions American Institute of Electrical Engineers*, 1944, "The Accuracy of Measurement of R. C. Cable Circuits in the Study of Transient Heat Flow."

In spite of the accuracy of measurement, the results obtained by the electrical method are only approximations. The method cannot take care of the effects of temperature on thermal conductivity and on heat transfer to the surface. Neither is the change of specific heat at the critical temperature of steel covered. However, the approximations are sufficiently close for most practical purposes, if judgment is used in the interpretation of the measurements.

Full-scale experiments in appropriately large furnaces furnish the most reliable information. Such tests require that holes be drilled in the solid that is to be heated, for the insertion of thermo-couples. In the heating of steel a number of such tests are from time to time published in the transactions of the Association of Iron and Steel Engineers. Again, judgment is needed in applying the results of full-scale tests, because test conditions may have differed from the conditions to which the results are to be applied.

appendix 2

Heat Transmission within a Furnace Enclosure

The following is a detailed example illustrating the method of Heilgenstaedt, used in determining the values of Tables 5 and 6 in Chapter 2.

As described in that chapter, this method consists in calculating the wall temperature by trial and error for given combinations of values of T_g (gas temperature), T_s (stock temperature), s (thickness of gaseous layer), and R (ratio of exposed stock area to refractory area). Throughout the calculation, the value of e (emissivity) is 0.9 for both stock and wall, and the coefficient of heat transfer by convection in the furnace is taken as 3.0 Btu per (sq ft, hr, F). Fuel is natural gas, with 9.5 per cent CO_2 and 19.0 per cent H_2O in the flue-gases, so that the partial pressures (p) of these gases are 0.095 and 0.190 respectively. By trial and error, the wall temperature is determined with which the calculated transfer of heat by radiation and convection from gas to wall equals the transfer of heat by radiation alone from wall to stock. Having determined the wall temperature (Table 6), the total heat transferred to the stock can be calculated (Table 5).

EXAMPLE.

Ratio R	0.5		
T_g	2400 F		
T_s	1700 F		
s	4 ft		
Estimated wall temperature, T_w	2045F	2040F	
Gas to wall by radiation			
k_1 from Fig. 37	132	131	Btu per (sq ft, hr, F)

ps for CO_2 = 0.095 × 4 ft	0.38	0.38	ft × atmosphere fraction
Emissivity CO_2 at gas temp. (Fig. 39)	0.087	0.087	
ps for H_2O = 0.190 × 4 ft	0.76	0.76	
Emissivity H_2O at gas temp. (Fig. 40)	0.130	0.130	
95% × the sum (page 41)e_1	0.21	0.21	
$k_1 \times 0.9$ (e, wall) $\times e_1(T_g - T_w) \times 1$ sq ft	8,850	8,950	Btu per (sq ft, hr)
Gas to wall by convection			
$3.0 \times (T_g - T_w)$	1,065	1,080	Btu per (sq ft, hr)
Gas to wall, total	9,915	10,030	Btu per (sq ft, hr)
Wall to stock by radiation			
k_2 from Fig. 37	86	85	
Emissivity CO_2 at wall temp	0.10	0.10	
Emissivity, H_2O at wall temp	0.13	0.13	
96% × sum e_2	0.22	0.22	
$k_2 \times (0.9 - e_2) \times R(T_w - T_s)$	10,050	9,850	
True wall temperature	2040 F		
Gas to stock by radiation			
k_3 from Fig. 37	110		
e_1	0.21		
$k_3 \times 0.9 \times e_1 \times (T_g - T_s) \times 1$ sq ft	14,600		
Gas to stock by convection			
$3.0 \times (T_g - T_s)$	2,100		
Wall to stock by radiation			
$9{,}850 \times 1/R$	19,700		
Total Btu per (sq ft, hr) to stock	36,400		
Coefficient of heat transfer $\dfrac{36{,}400}{2400 - 1700}$	52.0		

appendix **3**

Heat Losses from Walls

1. SURFACE HEAT TRANSFER

The heat-transfer coefficients shown in Fig. 79 are derived from the tests made by Wamsler[1] at the Laboratorium für technische Physik, München, Germany; by Langmuir[2] at Schenectady; by Heilman[3] at Mellon Institute; and by Griffith and Davis[4] at the National Physical Laboratory, Teddington, England.

The radiation coefficient (emissivity) as determined by Wamsler for rough, oxidized wrought-iron surfaces is 89 per cent of the absolute blackbody coefficient (see page 33), for rough cast-iron surfaces 90 per cent, and for rough white mortar surfaces 88 per cent of the blackbody coefficient. Other experimenters[2,5,6] have found lower values for the emissivity of similar surfaces, but Wamsler's results are confirmed by those of Heilman and are believed to be correct for the brick or metal surfaces of industrial furnace walls. Painting the surfaces has little effect on either radiation or convection losses, and the color of the surface has no influence whatever.[1] An exception is aluminum paint or "bronze," which reduces the radiation loss by about half, as long as it remains clean.

The convection coefficient for natural (not forced) convection varies approximately as the fourth root of the temperature difference between the surface and the surrounding air.[2,3,4] The evidence is conflicting as to whether the roughness of the surface affects the coefficient in the case of natural convection. The coefficient does, however, vary with the size of the surface and with its position. With the surface horizontal, facing upward, the convection loss is 10 to 29 per cent greater, and with the surface facing downward it is 33 to 50 per cent less than that of the vertical surface at the same temperature.[2,4] In small sizes, the natural-convection coefficient decreases approximately in inverse proportion to the fifth root of the height,[3] but for heights greater than 20 in. it remains constant.[4]

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[1] Wamsler, *Zeitschrift des Vereines deutscher Ingenieure,* 1911, page 599.

[2] Langmuir, *Transactions American Electrochemical Society,* Volume XXIII, 1913, page 299.

[3] Heilman, *Transactions American Society of Mechanical Engineers,* Volume 51, 1929, FSP 51-41.

[4] Griffith and Davis, *Engineering,* Volume 114, 1922, page 672.

[5] Schmidt, *Beiheft zum Gesundheits-Ingenieur,* Volume 20, 1927.

[6] Green, *Ceramic Society* (British) *Transactions,* Volume 25, 1926, page 383.

From the test data referred to, the following values have been selected as representing the average natural-convection coefficient for the exterior surfaces of industrial furnaces:

$$k_2 = 0.28\sqrt[4]{(T_0 - T_a)} \tag{44}$$

where k_2 = rate of heat transmission by conduction through the air film on the surface (see page 31) and convection, in still air, Btu per (sq ft, hr, F), from a surface at T_0 F to air at T_a F. The overall heat-transfer coefficient, in Btu per (sq ft, hr, F), then is

$$k = \frac{0.155\left[\left(\frac{T_0 + 460}{100}\right)^4 - \left(\frac{T_a + 460}{100}\right)^4\right]}{(T_0 - T_a)} + 0.28\sqrt[4]{(T_0 - T_a)} \tag{45}$$

including both radiation and convection and representing average values for furnace walls. From this expression and the related equation 14 on page 114, the curves of Figs. 79 and 80 were computed. They coincide almost exactly with the test results of Miketta[7] for the overall dissipation of heat from the outer, red-brick surface of a furnace wall. Other experimenters[8] have found losses as much as 12 per cent greater, but the test data seem erratic; it is probable that drafts of air are responsible.

2. CONDUCTIVITY OF MATERIALS

From the many tests that have been made of the heat conductivity of refractory materials,[9] the values shown by the curves of Fig. 357

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[7] Miketta, *Power,* 1927, page 127.

[8] Norton, *Fuels and Furnaces,* 1932, page 577.

[9] Beecher and Buckner, *Journal American Ceramic Society,* Volume 7, 1924, page 19; Clement and Egy, University of Illinois Experiment Station Bulletin 36, 1909; Dougill, Hodsman, and Cobb, *Iron & Coal Trades Review,* June 25, 1915, or *Journal Society Chemical Industry* (London), 1915, page 465; Boyd Dudley,

have been selected. With one exception, the curves shown apply to materials in use in the United States.

Only results obtained by the calorimeter method, under steady-state temperature conditions, have been shown; those derived from diffusivity determinations under varying temperature seem to be uniformly lower.

The multiplicity of coefficients is bewildering; variations in density, composition, and method of burning affect the conductivity. Data obtained from tests later than the third edition were not entered in Fig. 357, because they do not materially differ from the older data, and because they would add to the bewilderment.

In general, the conductivity of a given kind of brick decreases as the porosity increases, as shown by the Goerens-Gilles curves in Fig. 357, but no definite relation exists (Green, Esser).

In insulating materials, such as diatomaceous bricks, age appears to be a great factor. Experimenters have stated to the senior author that the conductivity of these materials is quite low when they are new and is considerably higher after they have been in use for some time.

This is true to some extent even for ordinary firebrick. The tests of Heyn, Bauer, and Wetzel showed that "underburned" firebricks—those that have not been exposed to very high temperatures in the process of manufacture—increased in conductivity as much as 20 per cent when heated to 2700 F. Green found that "semi-silica" brick which had been exposed to heat in a coke-oven wall for eighteen years had 10 per cent greater conductivity at 930 F than the original material, and 22 per cent greater at 2400 F.

In natural insulating bricks, which are of laminar structure, the conductivity for heat flow parallel to the layers (4½-in. way) is in some cases as much as 40 per cent higher than that for flow across the layers (2½-in. way).

Insulating refractories (lightweight firebricks, refractory insulating

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Transactions American Electrochemical Society, Volume XXVII, 1915, page 285; Esser, Salmang, and Schmidt-Ernsthausen, "Zur Kenntnis der Wärmeübertragung durch feurfeste Brennstoffe," Verlag des "Sprechsaal," 1930; Eucken and Laube, *Tonindustrie-Zeitung,* Volume 53, 1929, page 1599; Goerens and Gilles, *Mitteilungen aus dem eisenhüttenmännischen Institut der technischen Hochschule Aachen,* 1916, or *Ferrum,* Volume XII, no. 1, page 1; Green, *Ceramic Society* (British) *Transactions,* 1921, Volume 21, part 4, page 394; 1923, Volume 23, part 3, page 253; 1925, Volume 24, page 228; 1926, Volume 25, page 258; 1927, Volume 26, pages 159, 168; Griffiths, *Transactions Faraday Society,* Volume 12, 1917, page 193; Hartmann, Westmont, and Weinland, *Ceramic Age,* December 1928, page 225; *Transactions American Electrochemical Society,* Volume L, 1926, page 124; Heilman and Barnitt, *Proceedings Engineers Society of Western Pennsylvania,* Volume 47, 1931, page 349; Hersey and Butzler, Bureau of Mines Report 2564, April 1924; Heyn, Bauer, and Wetzel, *Untersuch. a. d. Kgl. Materialprüfungsamt,* Volume XXXII, 1914, Hefte 2 and 3; Norton, *Journal American Ceramic Society,* Volume 10, 1927, page 30; Takodoro, *Tohoku University Science Reports,* Volume 10, 1921, page 339; Van Rinsum, *Forschungsarbeiten auf dem Gebiete des Ingenieurwesens, Verein deutscher Ingenieure,* Heft 228, 1920.

bricks) are available for all temperatures that are encountered in industrial furnaces. To some extent, the thermal conductivity of these bricks is inversely proportional to their porosity and, therefore, to the

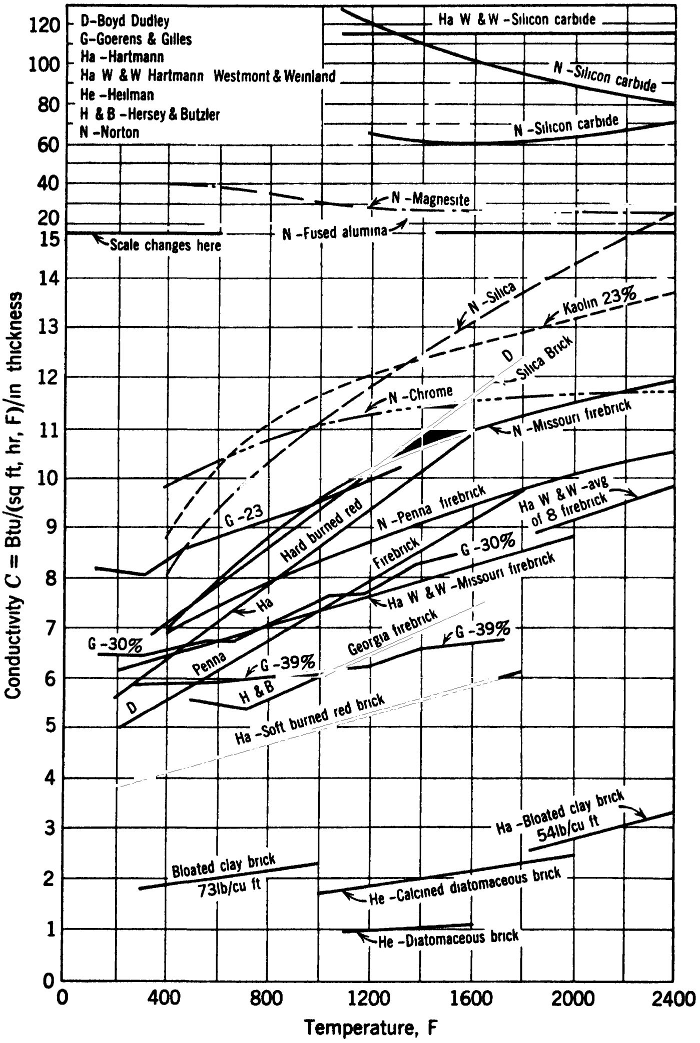

Fig. 357. Conductivity of refractory materials, plotted from test data.

overall density of the bricks. However, the effect of porosity is not an exact function, because it is not the total volume of pores that counts, but the number of pores in that volume. There are two reasons for this statement.

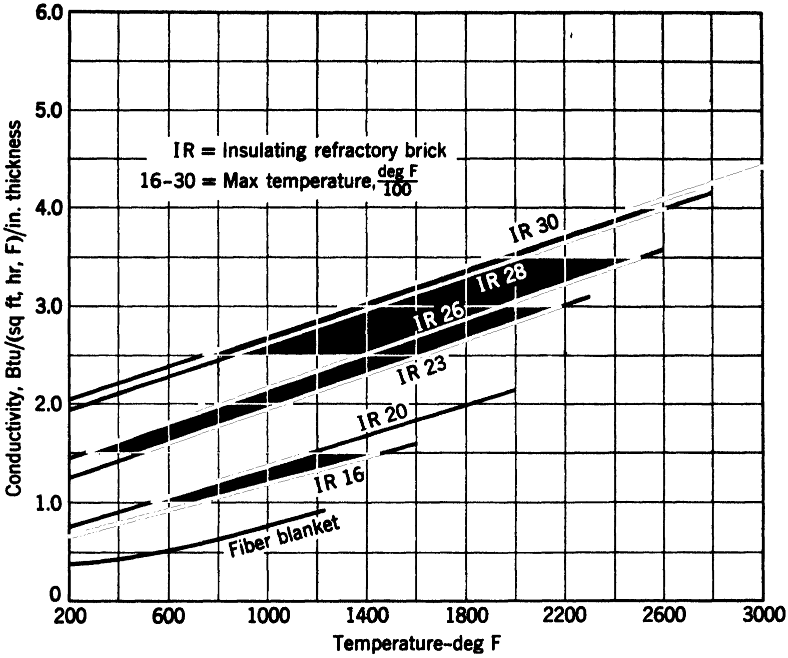

Fig. 358. Average conductivity of insulating materials.

1. The wall of each pore is a radiation screen, and
2. Heat-transmitting gas circulation in each pore is roughly proportional to the diameter of the pore.

It is manifestly impossible so to conduct the brick-making process that a predetermined number of pores having a predetermined diameter is obtained in each brick. For that reason, scientific tests of conductivity exhibit a wide spread of values. If all available test results were entered into a set of curves, the result would be an inextricable jumble. For that reason, only average values are given in Fig. 358.

3. GAS LEAKAGE THROUGH SAND SEALS

Furnaces with movable hearths, such as car-type furnaces and furnaces with rotating hearths, are equipped with sand seals. Hoods of annealing furnaces and also certain recuperator tubes rest in sand seals. Although sand seals are located at the bottom of furnaces, where the internal pressure is smallest, gas leakage through the seals is by no means inconsiderable and becomes of importance, if the furnace is filled with a prepared protective gas. Nevertheless, reliable information on sand seals is scarce. Lichterscheidt and Schmidt[10] found the following relation:

$$\frac{\text{Leakage, cu ft/hr}}{\text{Foot length}} = \frac{f \times \text{Pressure difference (inches of water)}}{\text{Depth of seal (inches)}}$$

The factor f was found to range between 2¼ and 3¼, varying with depth of immersion of blade and with fineness of sand or other filling.[11]

Thus, for a sand seal of 4-in. depth and a furnace pressure of 0.025 in. of water at the seal, the leakage (cold measure) ranges between 0.014 and 0.02 cu ft per hr and per ft of length of seal.

Approximately, the rate of leakage (referred to cold measure) is inversely proportional to the absolute temperature of the gas in the seal.

In some furnaces with rotating hearths, the sand seal was replaced by water seals. Such seals are gas tight but the water evaporates. The water level is automatically kept constant, but water evaporates into the furnace and increases the content of water vapor in the furnace atmosphere.

[10] *Gas Wärme* 2, 1953.

[11] The flow given by this equation is referred to 32 F. To express the flow in volume at higher temperatures, the factor must be multiplied by the rate of absolute temperatures.

appendix 4

Recuperator and Regenerator Calculations

1. USE OF LOGARITHMIC MEAN TEMPERATURE DIFFERENCE

If the heat-transfer coefficient and the specific heat were constant in all parts of the gas and air paths, the temperature distribution curves would be of an exponential form, and the mean temperature difference (see Fig. 149) in counterflow recuperators would be

$$T_g' - T_a' = \frac{(T_{1g} - T_{2a}) - (T_{2g} - T_{1a})}{\ln\left(\dfrac{T_{1g} - T_{2a}}{T_{2g} - T_{1a}}\right)} \tag{46}$$

T_{1g} being temperature of gases entering, T_{2g} temperature of gases leaving recuperator, T_{1a} temperature of air entering, and T_{2a} temperature of air leaving recuperator.

For parallel-flow recuperators, the same equation holds except that T_{1a} and T_{2a} are interchanged.

As stated in Chapter 6, the coefficient is by no means constant. On account of gaseous radiation, it is greater at the hot end than it is at the cold end. For that reason, calculation on the basis of the arithmetical mean temperature difference is, in almost all cases, just as accurate (or as inaccurate) as the use of the logarithmic mean temperature difference. Nevertheless, the "logarithmic mean temperature difference" as given by equation 46 is widely used as a basis for calculating recuperator performance. For that reason the alignment chart in Fig. 359 is offered,* by means of which the logarithmic mean can easily be found without

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* Due to Mr. David Moskovits, Carnegie Institute of Technology.

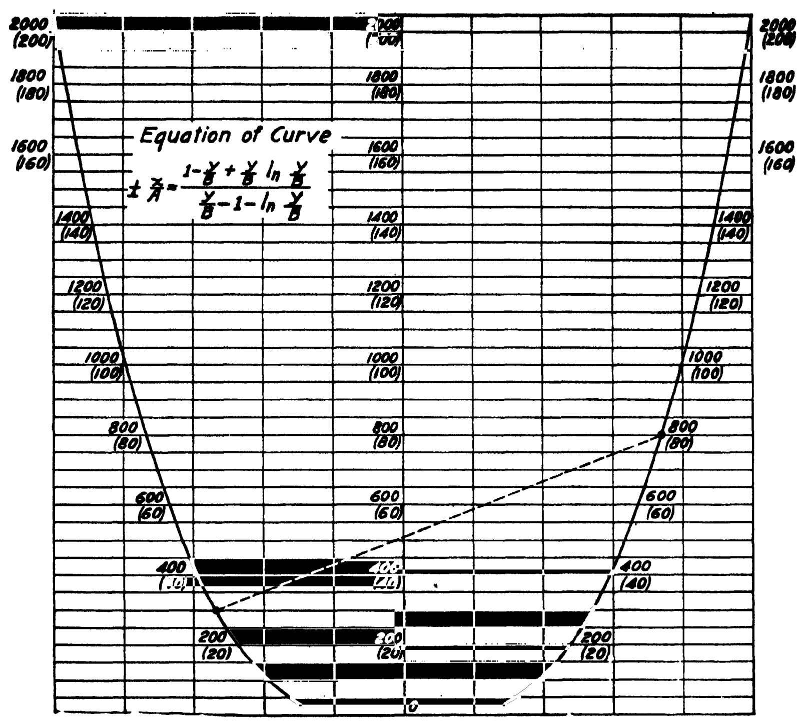

Fig. 359. Chart for determining logarithmic mean temperature difference. In the example shown by the dotted line, the logarithmic mean between 800 and 300 F is 508 F.

calculation. For example, if the gases enter a counterflow recuperator at 2200 F and leave at 900 F, while the air enters at 100 F and leaves at 1900 F, the temperature difference at the hot end is 300 F and at the cold end it is 800 F; by drawing a straight line (shown dotted in Fig. 359) from the 300 F point on the left branch of the curve to the 800 F point on the right branch, the intersection with the vertical center-line is obtained at 508 F, and this is the logarithmic mean temperature difference between the gas and air. (The arithmetical difference equals 550 F.)

2. RELATION BETWEEN HEATING SURFACE AND TEMPERATURE OF PREHEAT

It is sometimes desired to calculate the increase of preheat that could be obtained by deepening a checker or enlarging a recuperator.

For a given weight of air and gases, for constant heat-transfer coefficient, and for constant entering temperature of gas and air, the temperature relations vary as indicated in Fig. 360, if the recuperator is enlarged. The heavy curved line shows how the temperature to which the air is heated becomes greater as the surface is increased—the gain for a given increment becoming less, the greater the total area. The equation for final temperature of the air, based on logarithmic mean temperature difference, is

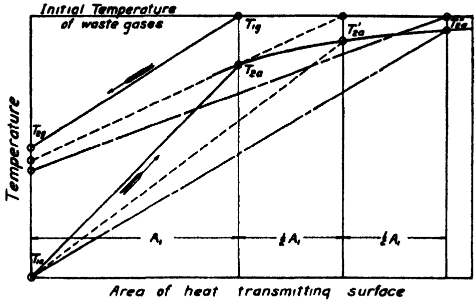

Fig. 360. Relation of preheat temperature to area of heat-transmitting surface.

$$T_{2a} = T_{1g} - \frac{(T_{1g} - T_{1a}) \times \left(1 - \dfrac{W_g \cdot c_g}{W_a \cdot c_a}\right)}{1 - \dfrac{W_g \cdot c_g}{W_a \cdot c_a} \times e^{-\frac{k \cdot A}{W_g \cdot c_g}\left(1 - \frac{W_g \cdot c_g}{W_a \cdot c_a}\right)}} \tag{47}$$

as derived by K. Rummel.* In this equation, W_g is weight of gases, and W_a equals weight of air, pounds per hour; c_g equals specific heat of gases, and c_a equals specific heat of air; k equals heat-transmission coefficient, Btu per (sq ft, hr, F); and A equals area of heat-transmitting surface, square feet. To obviate the difficulty of evaluating this cumbersome expression, the curves of Fig. 361 have been plotted; their resemblance to the heavy line of Fig. 360 will be evident. Abscissa, ordinate, and parameter are all dimensionless ratios. The following example will illustrate the use of these curves:

EXAMPLE. A producer-gasfired furnace using 150 cu ft of combustion air per min has an air checker 10 ft deep containing 12,800 sq ft of heating surface and heating the air to 1800 F. The gases, containing 40 per cent excess air, enter the checkers at 2300 F, 60 per cent of the gases going to the air checker. How much could the preheat temperature be raised by increasing the depth of the checkerwork from 10 to 13 ft?

The first step is to find what heat-transfer coefficient is obtained in the present checker.

---

* Melchett lecture, Institute of Fuel, London, 1931.

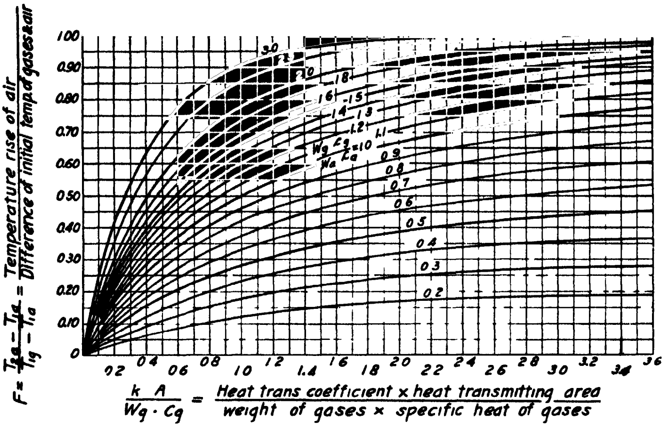

Fig. 361. Chart for determining temperature to which air is heated in a recuperator.

$$\text{Parameter} = \frac{\text{weight of gases} \times \text{specific heat of gases}}{\text{weight of air} \times \text{specific heat of air}} .$$

Weight of air = 150 × 0.0764 × 3600 = 41,300 lb per hr. From Fig. 116, weight of waste gases = 1.75 times the weight of air, or including 40 per cent excess air = 2.15 times. If 60 per cent of the gases go to the checker, W_g = 60% × 2.15 × 41,300 = 53,300 lb per hr. Specific heat of gases = approximately 0.31, of air = 0.25. Hence, to heat the air from 60 to 1800 F, the gases must cool $(1800 - 60) \times \frac{41{,}300 \times 0.25}{53{,}300 \times 0.31} = 1090$ F (neglecting wall losses), and their final temperature = 1210 F. Temperature difference at hot end = (2300 − 1800) = 500 F; at cold end = (1210 − 60) = 1150 F; arithmetical mean = 825 F; and, from Fig. 359, logarithmic mean = 780 F. Heat-transfer coefficient

$$k = \frac{41{,}300 \text{ lb per hr} \times 0.25 \text{ sp ht} \times (1800 - 60)}{12{,}800 \text{ sq ft area} \times 780 \text{ F}} = 1.80 \text{ Btu per (sq ft, hr, F)}$$

Figure 361 can now be used to find the preheat temperature for the deeper checker. If the brick setting is the same for the added checkerwork, the total heating surface would be 12,800 sq ft × $\frac{13}{10}$ = 16,650 sq ft.

$$\text{Abscissa in Fig. 361} = \frac{1.80 \times 16{,}650}{53{,}300 \times 0.31} = 1.81$$

$$\text{Parameter in Fig. 361} = \frac{53{,}300 \times 0.31}{41{,}300 \times 0.25} = 1.60$$

Factor *f* from that figure = 0.840

Temperature to which air is preheated

$$= T_{1a} + (T_{1g} - T_{1a})f = 60 + (2300 - 60) \times 0.84 = 1940 \text{ F}$$

Deepening the checker 30 per cent would therefore result in an increase of 140 F in preheat temperature. In these calculations it is assumed that the effect of increased wall losses can be neglected, and that the heat-transfer coefficient remains the same for the deepened checker as it was in the original checker. Neither of these assumptions is correct.

In spite of its formidable appearance, equation 47 cannot be expected to give the preheat temperature accurately because the heat-transfer coefficient is not constant. After obtaining a result from equation 47, the coefficient should be refigured for the temperature conditions corresponding to the result, in order to make sure that the value of k used is correct for those temperatures.

In practice, results obtained by increasing the depth of checkers have at times been disappointing. In one open-hearth furnace, the checkers were made several feet deeper, but the gain in air preheat temperature was only a few degrees, or less than the accuracy of measurement. This was due chiefly to the fact that the heating surface originally present was sufficient to heat the air to within about 150 F of the temperature of the gases entering the regenerator, the conditions corresponding to points near the right-hand end of Figs. 360 and 361, where the curves have flattened out; and the small gain in preheat ideally obtainable by a moderate increase of heating surface was wiped out by the increase of wall losses from the checker chambers, and by increase of air infiltration.

3. EFFECT OF LEAKAGE IN RECUPERATORS

On the basis of uniform rate of leakage per foot of travel of the air and gases, the effect of such leakage in reducing the preheat temperature has been analyzed by W. Heiligenstaedt.* The resulting equation is

$$T_{a_2}\left[1+\frac{\dfrac{W_a\cdot c_a}{f\cdot W_g\cdot c_g}}{1+n\dfrac{W_a}{W_g}}+\frac{W_a\cdot c_a}{f\cdot k\cdot A}\cdot(2+n)\right]=T_{g_1}\left[1+\frac{1}{1+n\dfrac{W_a}{W_g}}\right]$$
$$+\,T_{a_1}\left[\frac{W_a\cdot c_a}{f\cdot W_g\cdot c_g}\cdot\frac{(1+n)}{\left(1+n\dfrac{W_a}{W_g}\right)}+\frac{W_a\cdot c_a}{f\cdot k\cdot A}\cdot(2+n)-1\right]\qquad(48)$$

T_{a_2}, as before, is the air preheat temperature. The wall loss has been taken care of by the factor f, which is the "efficiency" of the recuperator or the ratio $\dfrac{\text{Heat input to air}}{\text{Heat abstracted from gases}}$, and n is the ratio $\dfrac{\text{Leakage air weight}}{\text{Weight of air delivered by the recuperator}}$. The other symbols have the same meaning as previously. The equation is based on the arithmetical mean temperature difference, which leads to practically no error because in leaky recuperators the preheat temperature never approaches the gas temperature.

---

* "Regeneratoren, Rekuperatoren, Winderhitzer," Spamer, Leipzig, 1931.

Heiligenstaedt showed that, if the heat-transfer coefficient k remained constant, the preheat temperature would be reduced about 40 per cent by the leakage of as much air as is delivered by the recuperator ($n = 1$). Actually, the dilution reduces gas radiation, k decreases, and the loss of preheat temperature is even greater than calculated.

Depending on the relative pressures, flue gases may leak into the air passages or air may leak into the gas passages. In the latter case, the fan may be unable to supply enough air to make up for the leakage: first, because a greater volume is required; and second, because additional pressure is required to compensate for the increased pressure drop due to this larger volume. However, this additional pressure is seldom available because the fan motor either runs at constant speed or, even if variable, must usually be operated at maximum speed to produce even as much pressure as is required without air leakage.

4. EFFECT OF WALL LOSS IN REGENERATORS AND RECUPERATORS

Regenerators will be considered first. If H is the wall loss in Btu per hour from each chamber, then equation 23 becomes

$$W_g \cdot c_g \cdot (T_{1g} - T_{2g}) = W_a \cdot c_a \cdot (T_{2a} - T_{1a}) + 2H = A \cdot k \cdot (T_g' - T_a') + H \quad (49)$$

the other symbols having the same meanings as in Chapter 6 and Fig. 149. The reason why the $2H$ appears in the middle term is that losses are occurring from the walls of both regenerator chambers simultaneously, while heat is being transmitted to the combustion air in only one chamber at a time.

If A_o is the area of the walls of each checker chamber and k_o is the coefficient of heat passage through these walls in Btu per (sq ft, hr, F), then

$$H = A_o \cdot k_o \cdot \left(\frac{T_{1a} + T_{2a} + T_{1g} + T_{2g}}{4}\right) \text{ approximately}$$

Substituting this expression and the arithmetical mean temperature difference $(T_g' - T_a') = \left(\dfrac{T_{1g} + T_{2g} - T_{1a} - T_{2a}}{2}\right)$ in equation 49, and simplifying, we obtain

$$T_{2g} = \frac{T_{1g}\cdot\left(W_g\cdot c_g - \dfrac{A_o\cdot k_o}{2}\right) + T'_{1a}\cdot\left(W_a\cdot c_a - \dfrac{A_o\cdot k_o}{2}\right) - T_{2a}\cdot\left(W_a\cdot c_a + \dfrac{A_o\cdot k_o}{2}\right)}{\left(W_g\cdot c_g + \dfrac{A_o\cdot k_o}{2}\right)} \tag{50}$$

and

$$k = \frac{A_o\cdot k_o}{2A} \times \left[\frac{(T_{1g} + T_{2g} + T_{1a} + T_{2a}) + W_a\cdot c_a\cdot(T_{2a} - T_{1a})}{(T_{1g} + T_{2g} - T_{1a} - T_{2a})}\right] \tag{51}$$

from which k can be calculated if the other values are known.

Combining equations 49 and 50, we obtain

$$T_{2a} = \frac{\left\{\left(\dfrac{Ak}{2} - \dfrac{A_ok_o}{4}\right)\left[\dfrac{T_{1g}\left(W_gc_g - \dfrac{A_ok_o}{2}\right) + T_{1a}\left(W_ac_a - \dfrac{A_ok_o}{2}\right)}{\left(W_gc_g + \dfrac{A_ok_o}{2}\right)}\right] + T_{1g}\left(\dfrac{Ak}{2} - \dfrac{A_ok_o}{4}\right) - T_{1a}\left(\dfrac{Ak}{2} + \dfrac{A_ok_o}{4} - W_ac_a\right)\right\}}{\dfrac{\left(W_ac_a + \dfrac{A_ok_o}{2}\right)\cdot\left(\dfrac{Ak}{2} - \dfrac{A_ok_o}{4}\right)}{\left(W_gc_g + \dfrac{A_ok_o}{2}\right)} + \dfrac{Ak}{2} + \dfrac{A_ok_o}{4} + W_a\cdot c_a} \tag{52}$$

from which the final temperature to which the air is preheated can be found if k is known.

The following example will illustrate the use of this equation. In a regenerative furnace, each checker has 27,900 sq ft of heating surface A, and the area of the walls of the chamber, 18 in. thick, is 4500 sq ft, A_o; 33,500 lb of air per hr, W_a, enter at 200 F, T_{1a}. The waste gases, amounting to 37,000 lb per hr, W_g, enter at 2300 F, T_{1g}. To what temperature will the air be heated, if $k = 1.5$?

From Fig. 86, the coefficient k_o is figured as 0.36 Btu per (sq ft, hr, F); $c_a = 0.265$, $c_g = 0.31$.

$$T_{2a}=\frac{\left\{\begin{array}{c}\left(\frac{27{,}900\times1.5}{2}-\frac{4500\times0.36}{4}\right)\times\\ \left[\frac{2300\left(37{,}000\times0.31-\frac{4500\times0.36}{2}\right)+200\left(33{,}500\times0.265-\frac{4500\times0.36}{2}\right)}{\left(37{,}000\times0.31+\frac{4500\times0.36}{2}\right)}\right]\\ +2300\left(\frac{27{,}900\times1.5}{2}-\frac{4500\times0.36}{2}\right)\\ -200\left(\frac{27{,}900\times1.5}{2}+\frac{4500\times0.36}{4}-33{,}500\times0.265\right)\end{array}\right\}}{\left\{\begin{array}{c}\frac{\left(33{,}500\times0.265+\frac{4500\times0.36}{2}\right)\cdot\left(\frac{27{,}900\times1.5}{2}-\frac{4500\times0.36}{4}\right)}{\left(37{,}000\times0.31+\frac{4500\times0.36}{2}\right)}\\ +\frac{27{,}900\times1.5}{2}+\frac{4500\times0.36}{4}+33{,}500\times0.265\end{array}\right\}}$$

whence

$$T_{2a} = 1895\text{ F}$$

If there were no wall losses, the $A_o \cdot k_o$ terms would disappear, and T_{2a} would be 2110 F. The wall losses therefore reduce the final air temperature by 215 F in this case. The loss of temperature would be greater in a small regenerator, because for similar shapes of chamber the external area is proportional to the square of the linear dimensions, but the heating surface varies as the cube of the linear dimension; furthermore, the walls are usually thinner in the smaller chambers. Roughly, the wall loss may be said to be inversely proportional to the area of heating surface in the checker; it varies, however, with the type of wall construction.

Turning next to recuperators, we find that the wall loss depends on the arrangement. When the gases are outside and the air inside the tubes, the equation becomes

$$T_{2a}=\frac{T_{1a}\left(\frac{W_a\cdot c_a}{W_g\cdot c_g}-\frac{A_o\cdot k_o}{2W_g\cdot c_g}+\frac{2W_a\cdot c_a}{k\cdot A}\right)+2\cdot T_{1_g}-T_{1a}}{\frac{2W_a\cdot c_a}{k\cdot A}+1+\frac{W_a\cdot c_a}{W_g\cdot c_g}+\frac{A_o\cdot k_o}{2W_g\cdot c_g}} \tag{53}$$

and from it the effect of wall losses can be determined.

5. PRESSURE DROP AND HEAT TRANSFER

The heat-transfer coefficient for air or non-radiating gases flowing in tubes is approximately as given by the formula*

---

* The exponents for both velocity and diameter, as found by different experimenters, vary somewhat, but the difference in the final result is negligible. The coefficients in equations 54 and 55 take care of an average viscosity.

$$k = \frac{4.46(c'q)^{0.8}}{D^{0.25}} \tag{54}$$

where k = Btu per (sq ft, hr, F), c' = actual gas velocity, feet per second, and q = gas density, pounds per cubic foot; D = tube diameter, inches. If instead of the actual velocity at any place along the tube, the equivalent velocity c at 60 F is substituted, then $q = 0.0764$, and the equation becomes $k = 0.57 \times \frac{c^{0.8}}{D^{0.25}}$

The pressure drop caused by friction is given by Fig. 300 of Chapter 8, but it can also be approximately expressed by the equation

$$(p_1 - p_2) = 3.35 \times 10^{-7} \cdot \frac{T \cdot l \cdot c^{1.85}}{D^{1.27}} \tag{55}$$

where $(p_1 - p_2)$ = pressure drop, inches of water, T = absolute temperature of gases, degrees Fahrenheit, and l = length of tube, feet, D = diameter, inches.

If this were the only pressure drop, and the resistance on the air side the only resistance to heat flow, the relation between pressure drop and size of recuperator could be developed as follows: Area $= n \cdot \pi \cdot \frac{D}{12} \cdot l$, with n = number of tubes.

Hence

$$k \cdot A = 0.57 \cdot n \cdot \frac{\pi}{12} D \cdot l \cdot \frac{c^{0.8}}{D^{0.25}} = 0.15 \cdot n \cdot l \cdot D^{0.75} \cdot c^{0.8}$$

and

$$(p_1 - p_2) = 2.23 \times 10^{-6} \cdot \frac{T \cdot k \cdot A}{n} \cdot \frac{c^{1.05}}{D^{2.02}}$$

or very nearly

$$2.2 \times 10^{-6} \cdot \frac{T \cdot k \cdot A}{n} \cdot \frac{c}{D^2}$$

Also, if Q = quantity of air in cubic feet per second at 60 F,

$$c = \frac{Q \times 144}{\frac{\pi}{4} D^2 \cdot n}$$

Finally,

$$(p_1 - p_2)_1 = 4.03 \times 10^{-4} \cdot (k \cdot A) \cdot \frac{Q \cdot T_a'}{n^2 \cdot D^4}$$

T_a' is *absolute* average temperature. For given recuperator conditions, Q, T_a', and $k \cdot A$ are fixed, the last two being determined by the temperature conditions. Hence we may say that the pressure drop is inversely proportional to (number of tubes times square of diameter)^2; and, for any allowable pressure drop, the number and size of tubes could be found directly.

In the actual recuperator, however, there are two other factors to be considered. First, the overall coefficient of heat transfer is not the k given by the above formula, but, as shown in Chapter 6, it is $k = \dfrac{1}{1/k_a + 1/k_w + 1/k_g}$, and the last two terms are not affected by what happens on the air side. Second, additional pressure drop is required to give the air the necessary velocity at the entrance of the tube, or

$$(p_1 - p_2)_2 = 7.80 \times 10^{-7} \cdot T_{1a} \cdot c^2$$

and also to accelerate it as it becomes heated in the tube, or

$$(p_1 - p_2)_3 = 4.39 \times 10^{-7} \cdot (T_{2a} - T_{1a}) \cdot c^2 \tag{56}$$

where c = velocity at 60 F in feet per second. All temperatures are absolute. The total pressure drop is the sum of the three terms: namely, friction, velocity head at entrance, and acceleration in tube. Since it is impossible to reduce these complicated sums to a simple form, values for Fig. 152 were found by taking numerical examples and, for each value of c, figuring the overall coefficient k and the total pressure drop. For other dimensions of passages, the same procedure can be followed, using the above formulas.

The question often arises as to whether devices for producing additional turbulence, such as staggered tubes, "corebusters," whirlers, etc., are of advantage. Few data are available from which to judge. It has been found, however, that staggered tubes (which produce turbulence) result in a higher ratio $\dfrac{\text{Heat transfer}}{\text{Pressure drop}}$ than "in-line" tubes do, if $\dfrac{\text{Tube spacing, center to center}}{\text{Tube diameter}}$ is smaller than 1.5. For wider spacing, the ratio of heat transfer to pressure drop is greater for "in-line" tubes than it is for staggered tubes.

Corebusters are of two types. Those of one type produce a whirl and throw the otherwise inactive core to the surface of the tube. Those of the other type fill the core and force all of the gas or air to flow through an annulus near the surface of the tube. Some misleading statements have been published about the advantages and disadvantages of whirlers. One of them follows: "Whirlers multiply the heat transfer by six, at the expense of multiplying the pressure drop by 200, whereas, without whirlers, the same increase in heat transfer can be obtained with 60 times the original pressure drop, simply by increasing the velocity." This statement disregards the influence of the absolute size of tube diameter and of the ratio of tube length to tube diameter. For a given pressure drop, a greater number of tubes of smaller diameter increases the heat transfer much more than whirlers do. Either whirlers or smooth corefillers are indicated, whenever small long tubes are not available, as for instance in the case of silicon carbide tubes, or if the cost per square foot of heating surface grows rapidly for tubes of small diameter. Cylindrical corefillers furnish secondary heating surface. They are useful as inserts in large short tubes of expensive material, particularly if they can be made of a cheaper material than that of the tubes.

appendix 5

Properties of Materials That Are Seldom Heated in Industrial Furnaces

Some materials that are not included in Fig. 26 are occasionally heated in industrial furnaces. These materials are either the charge (heating stock, ware) or else auxiliary substances that are used in connection with heating the charge. The data were taken, with permission, from the "North American Combustion Handbook." In that book the original sources of information are not given.

Material	Density, lb/cu ft	Temp. Range, F	Specific Heat, Btu/lb, deg F	Melting (fusion) Point, deg F	Latent Heat, Btu/lb	
					of Fusion	of Vaporization
Antimony (Sb)	422	392	0.052	1166	68.9	703
Arsenic (gray)	358	32–212	0.0822			
Asbestos	124–174	32–212	0.20			
Barium (Ba)	218	−121 + 68	0.068	1562		1120
Beryllium	113.6		0.52	2732		
Boron (B)	152	32–212	0.307	3992–4532	570	
Cadmium (Cd)	540	212	0.057	608.6	23.8	409
Carbon (C) graphite	138	52	0.160	6332		
Charcoal	18–38	75	0.165–0.25			
Chromium (Cr)	449	212	0.1121	2929		
Cobalt (Co)	556	932	0.1542	2723	115.2	
Columbium (Cb)	535		0.065	4380		
Constantan		32–212	0.098			
Gallium (Ga)	367	54–235	0.080	86.18		
Germanium (Ge)	335	32–212	0.0737	1756.4		
Glass	144-187	32–212	0.15–0.23	2190		
Molybdenum (Mo)	637	68–212	0.0647	4595	126	
Monel metal	550	60–2415	0.129	2415	117.4	
Osmium (Os)	1402	68–208	0.0311			
Palladium (Pe)	749	32–2309	0.0714	2820	64.6	
Platinum (Pt)	1335	68–2372	0.0359	3191	48.96	

Material	Density, lb/cu ft	Temp. Range, F	Specific Heat, Btu/lb, deg F	Melting (fusion) Point, deg F	Latent Heat, Btu/lb	
					of Fusion	of Vaporization
Pyrex		68–212	0.196			
Rhodium	777	50–207	0.058	3542		
Rubidium	95.5	32	0.0802	100.4		
Sand	162	59–212	0.195			
Selenium (Se)	300	−306 + 64	0.068	422.6–428		
Silicon (Si)	145	135	0.1833	2588	607	
Sodium (Na)	60.6	−301 + 68	0.253	207.5	49.3	
Solder (Pb and Sn)	580		0.040–0.051	361–594	11.6–30.6	
Tantalum (Ta)	1035	2552	0.043	5252		
Tellurium (Te)	389	59–212	0.0483	845.6	13.1	
Thallium (Tl)	740	68–212	0.0326	575.6		
Thorium (Th)	699	32–212	0.0276	3350		
Titanium (Ti)	283	32–212	0.1125	3263		
Tungsten (W)	1202	32–212	0.0336		79	
Uranium (U)	1167	32–208	0.028	3344		
Vanadium	375	32–212	0.1153	3128		

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